US012077209B1

(12) **United States Patent**
Bonny et al.

(10) **Patent No.: US 12,077,209 B1**
(45) **Date of Patent: Sep. 3, 2024**

(54) **MODULAR DRIVE SYSTEM**

(71) Applicant: **Hydro-Gear Limited Partnership**, Sullivan, IL (US)

(72) Inventors: **Nathan W. Bonny**, Shelbyville, IL (US); **Raymond Hauser**, Franklin, TN (US)

(73) Assignee: **Hydro-Gear Limited Partnership**, Sullivan, IL (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 250 days.

(21) Appl. No.: **17/446,274**

(22) Filed: **Aug. 27, 2021**

Related U.S. Application Data

(60) Continuation of application No. 15/985,335, filed on May 21, 2018, now abandoned, which is a division of application No. 14/462,868, filed on Aug. 19, 2014, now abandoned, which is a continuation of application No. 13/732,928, filed on Jan. 2, 2013, now Pat. No. 9,194,473, which is a continuation-in-part of application No. 12/825,038, filed on Jun. 28, 2010, now Pat. No. 8,393,236.

(60) Provisional application No. 61/258,764, filed on Nov. 6, 2009, provisional application No. 61/243,097, filed on Sep. 16, 2009, provisional application No. 61/220,795, filed on Jun. 26, 2009.

(51) **Int. Cl.**

B62D 11/12	(2006.01)
A01D 34/00	(2006.01)
B60K 17/344	(2006.01)

(52) **U.S. Cl.**
CPC ***B62D 11/12*** (2013.01); ***A01D 34/00*** (2013.01); ***B60K 17/344*** (2013.01)

(58) **Field of Classification Search**
CPC B62D 11/12; B62D 11/08; A01D 34/00; B60K 17/344; B60K 17/24
See application file for complete search history.

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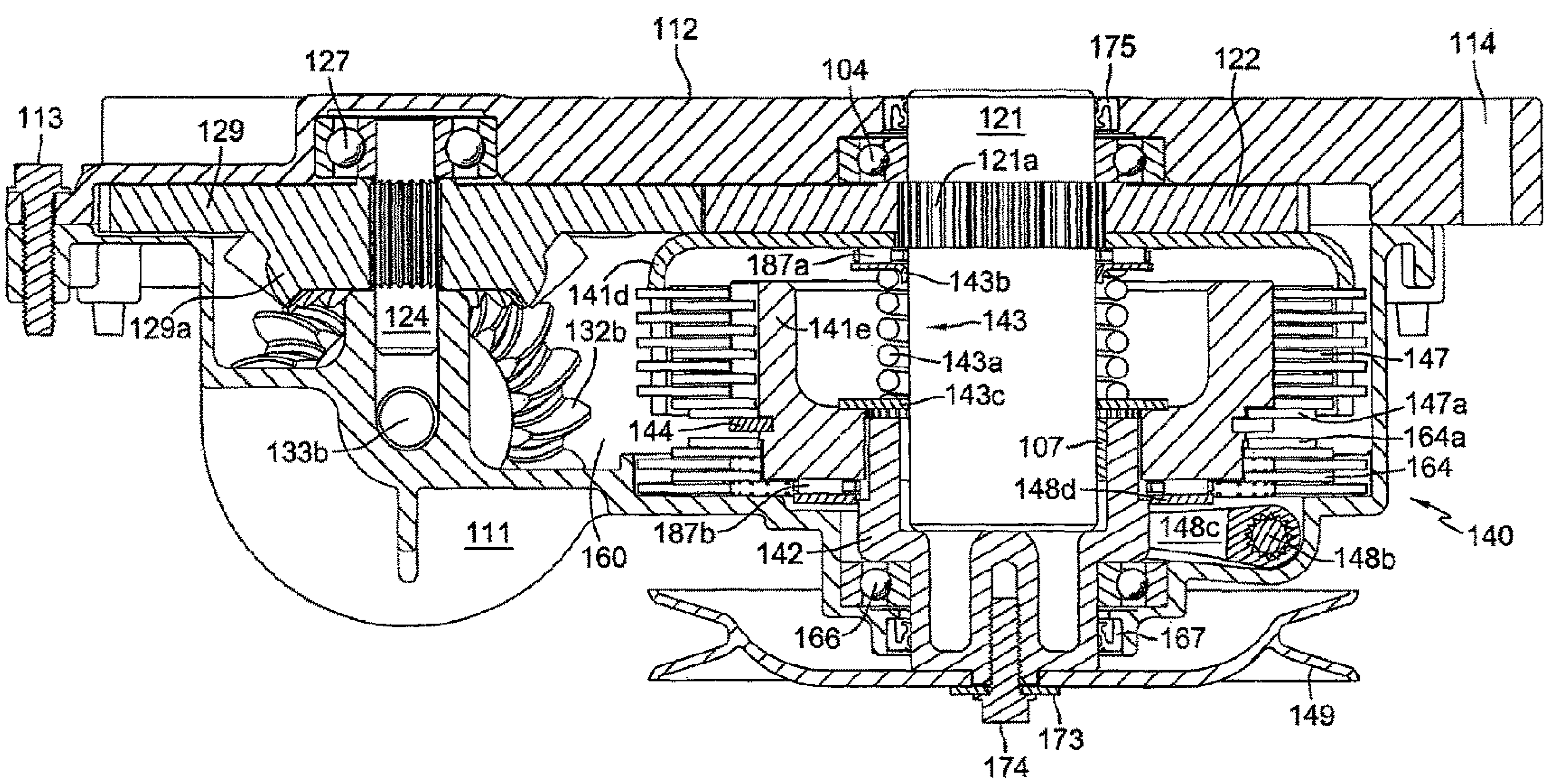
Primary Examiner — Timothy Wilhelm
(74) *Attorney, Agent, or Firm* — Neal, Gerber & Eisenberg LLP

(57) **ABSTRACT**

A vehicle having a prime mover and a central gear box mounted to a frame. First and second transaxles are driven by the central gear box. Each transaxle includes an output axle having a completely splined external portion and a threaded end for engagement with a driven wheel, and the completely splined portion of each output axle is disposed in one of the driven wheels of the vehicle. A cross brace may be mounted to the two transaxle housings for rigidity.

21 Claims, 49 Drawing Sheets



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U.S. Appl. No. 13/732,928, filed Jan. 2, 2013; and.
U.S. Appl. No. 12/825,038, filed Jun. 28, 2010.

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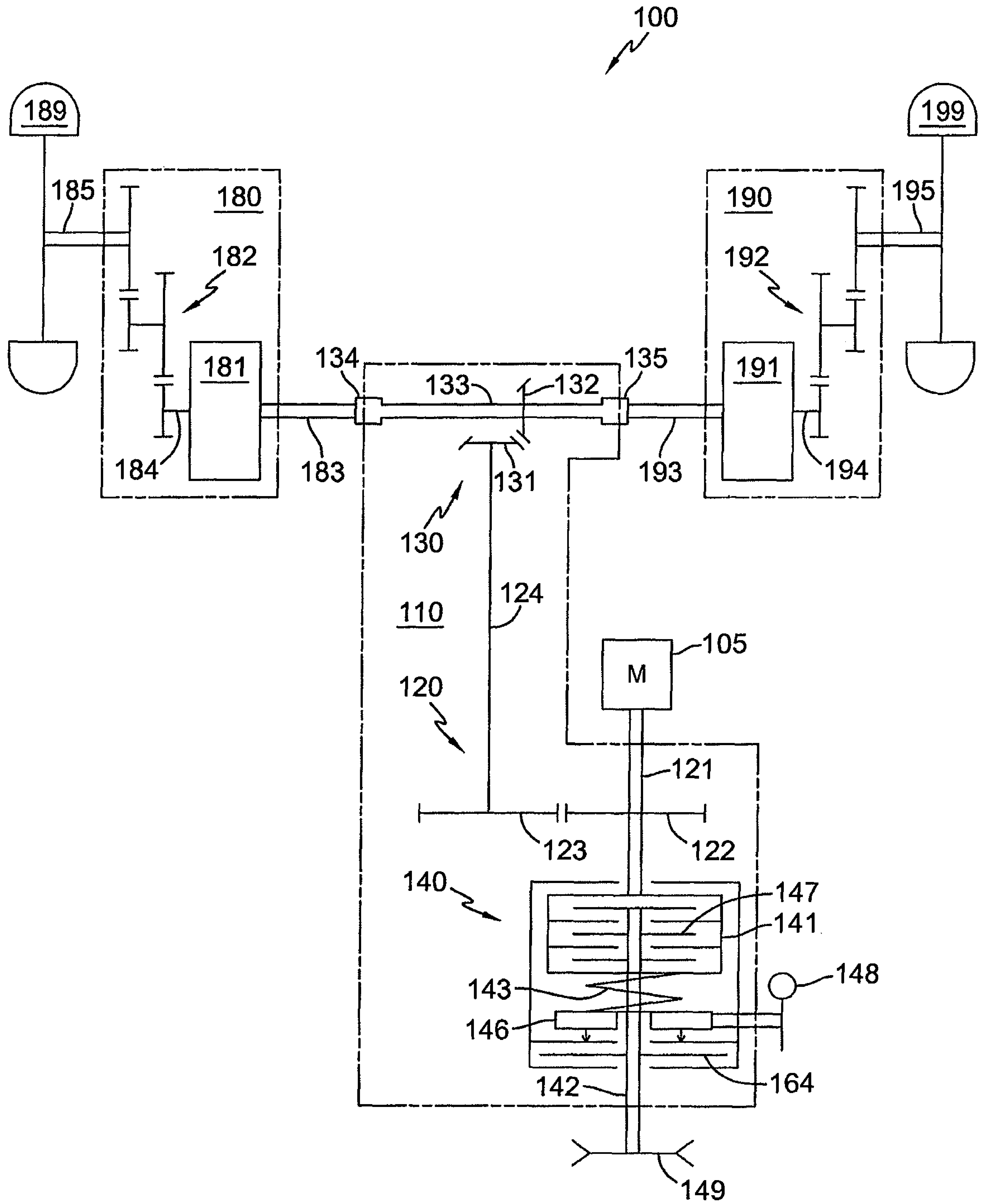
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FIG. 1

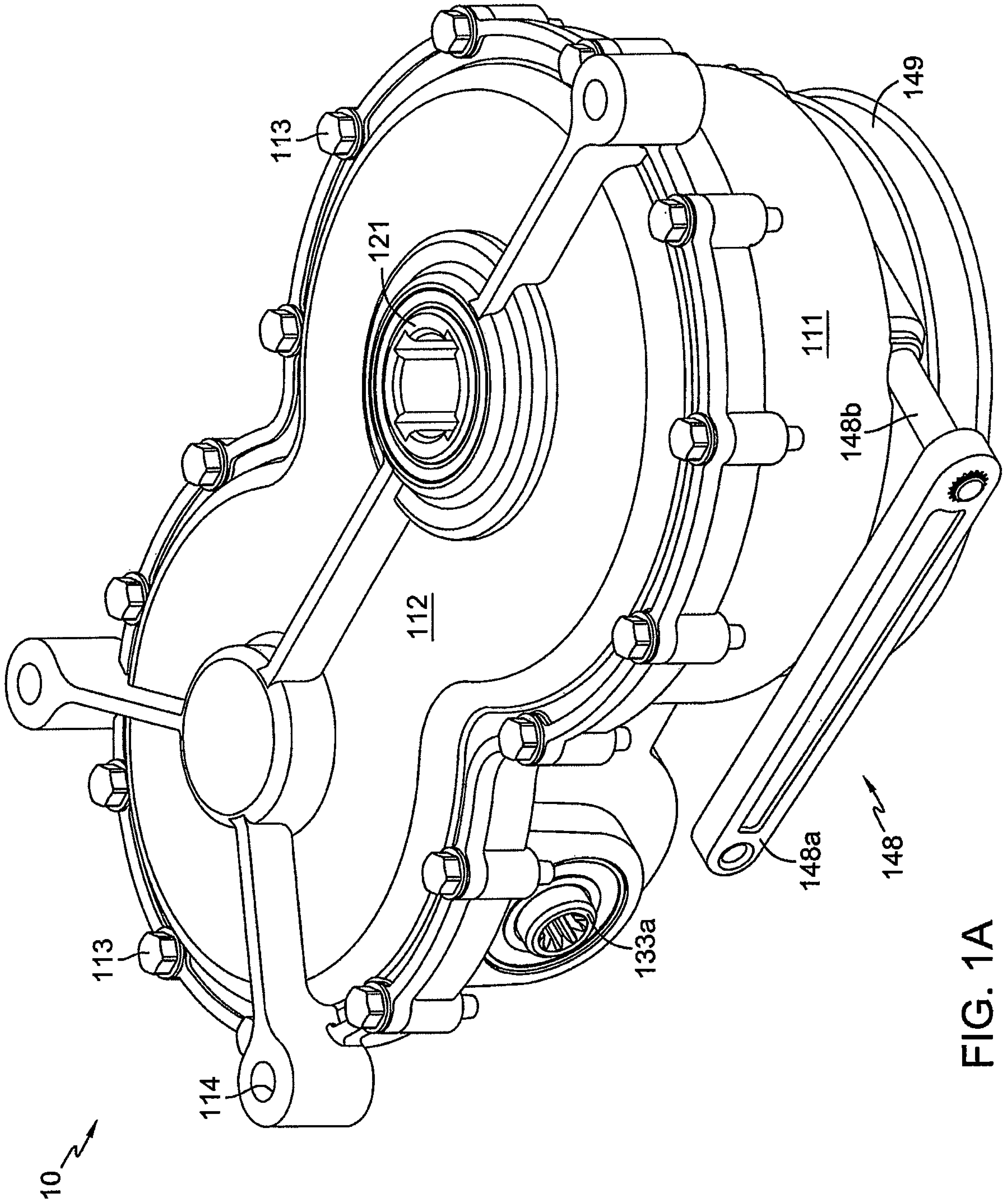


FIG. 1A

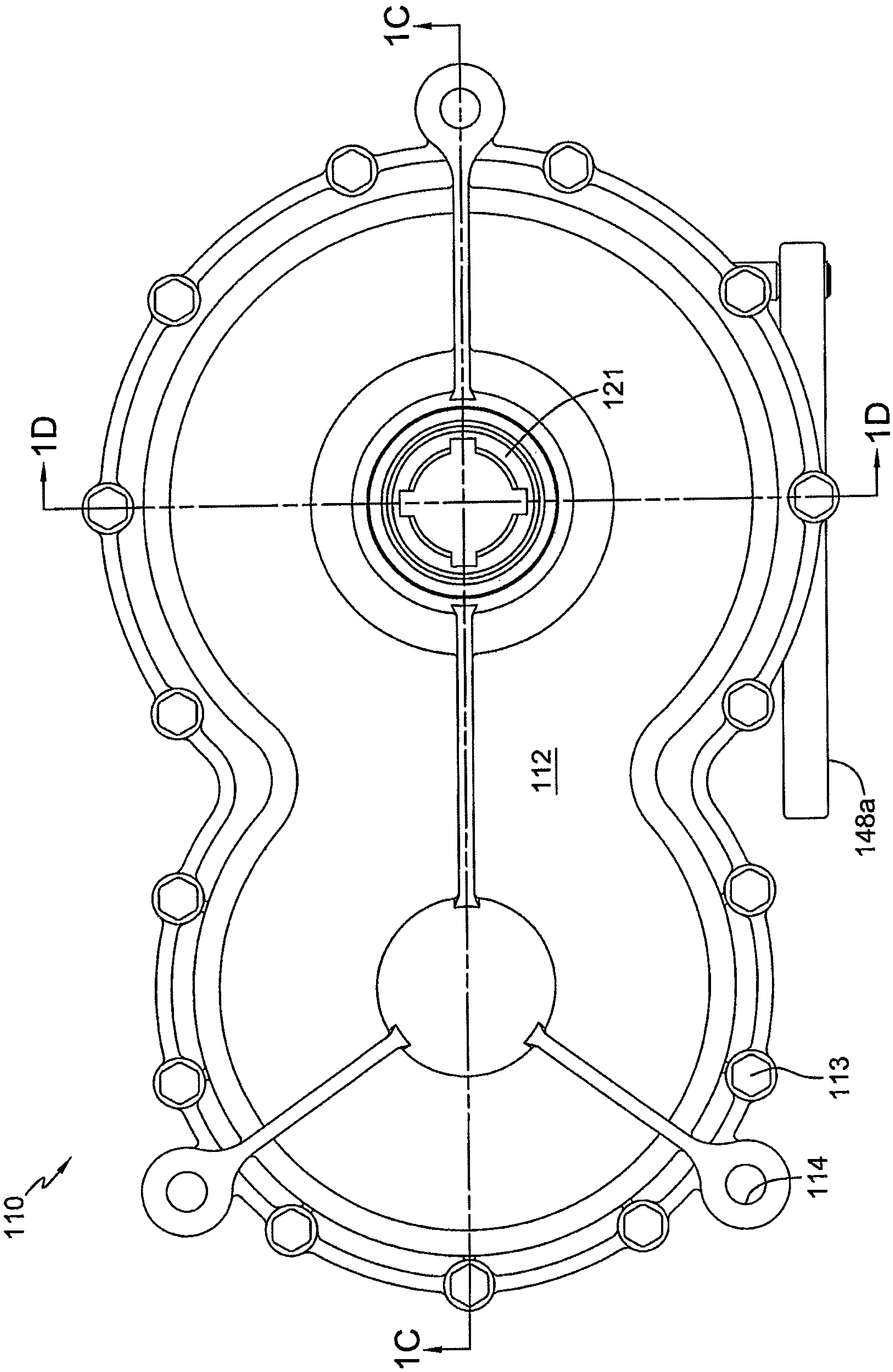
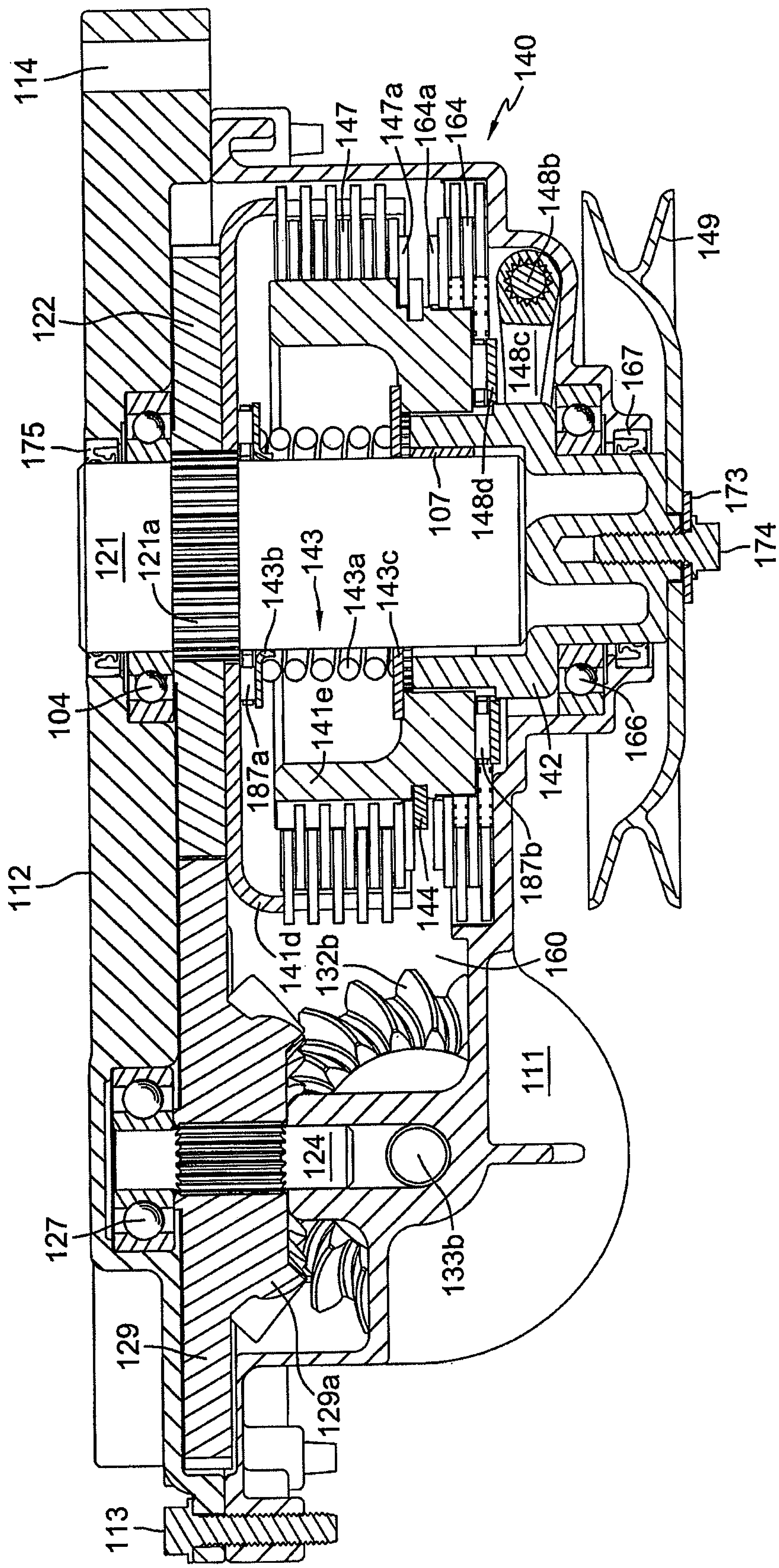


FIG. 1B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143a
143c
174
104
187a
141e
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144
187b
141d
132b
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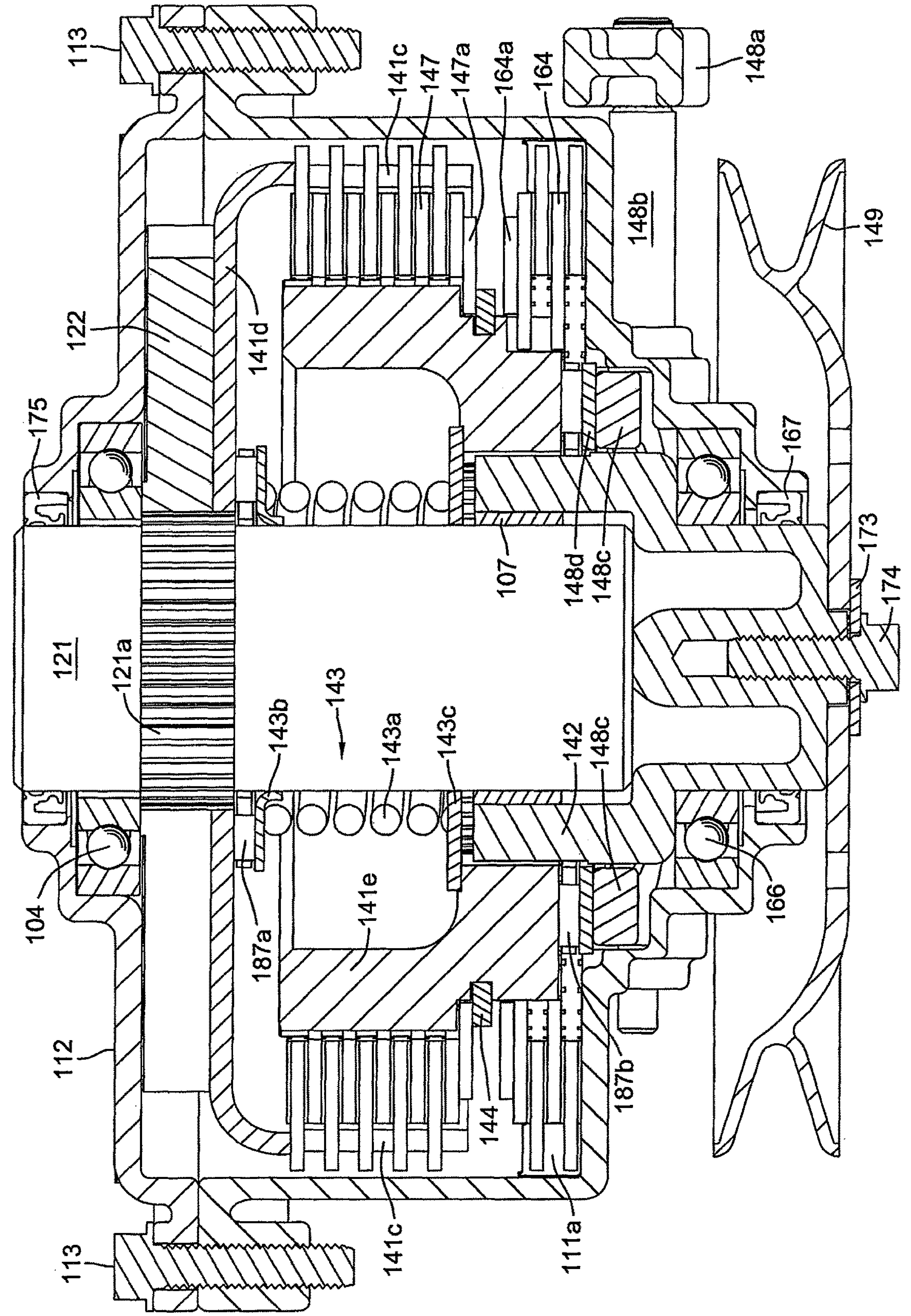
111
124
127
133b
129
129a
113

FIG. 1C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121a
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148c
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112
187b
144
141c
111a
113

FIG. 1D

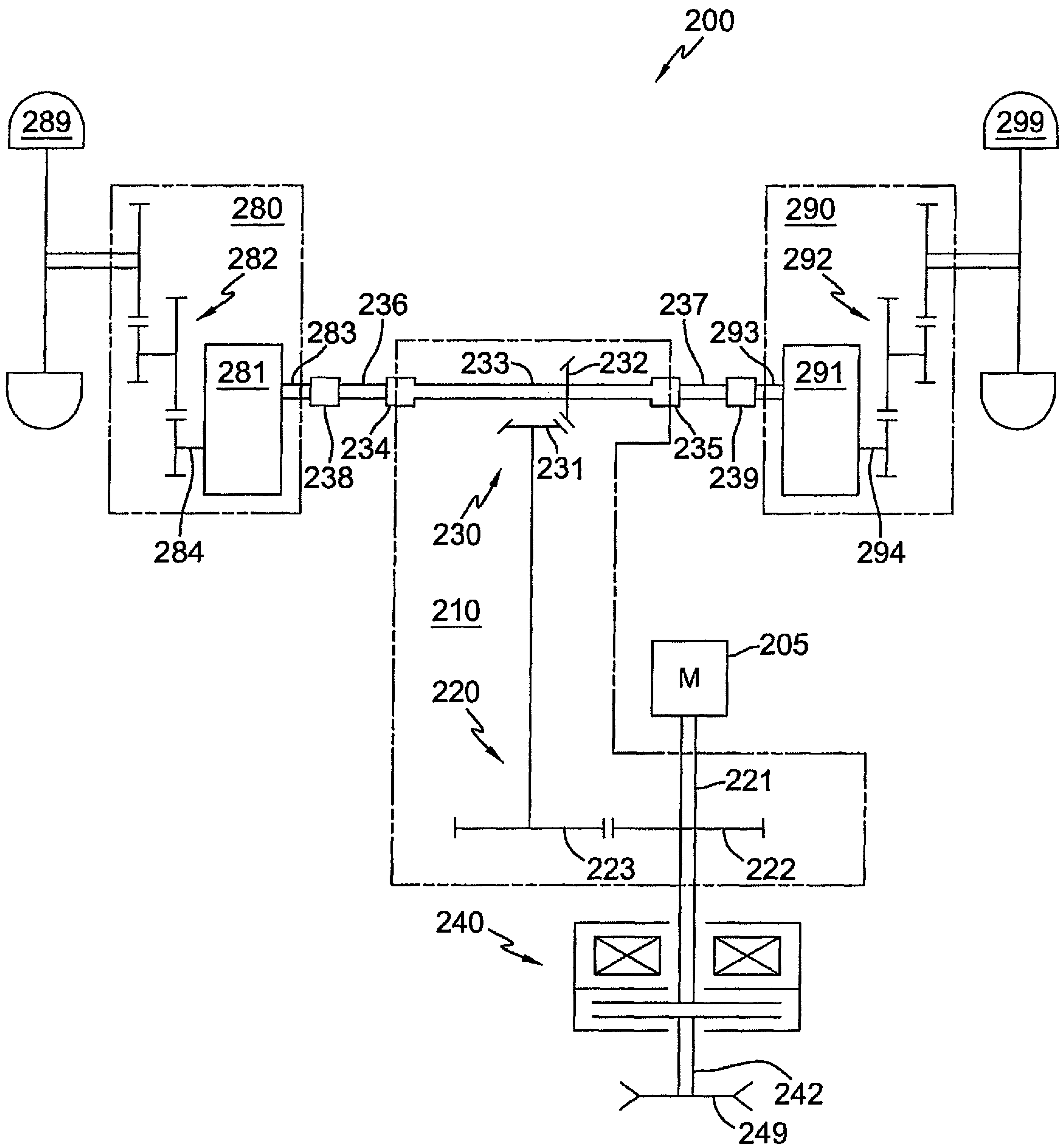
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FIG. 2

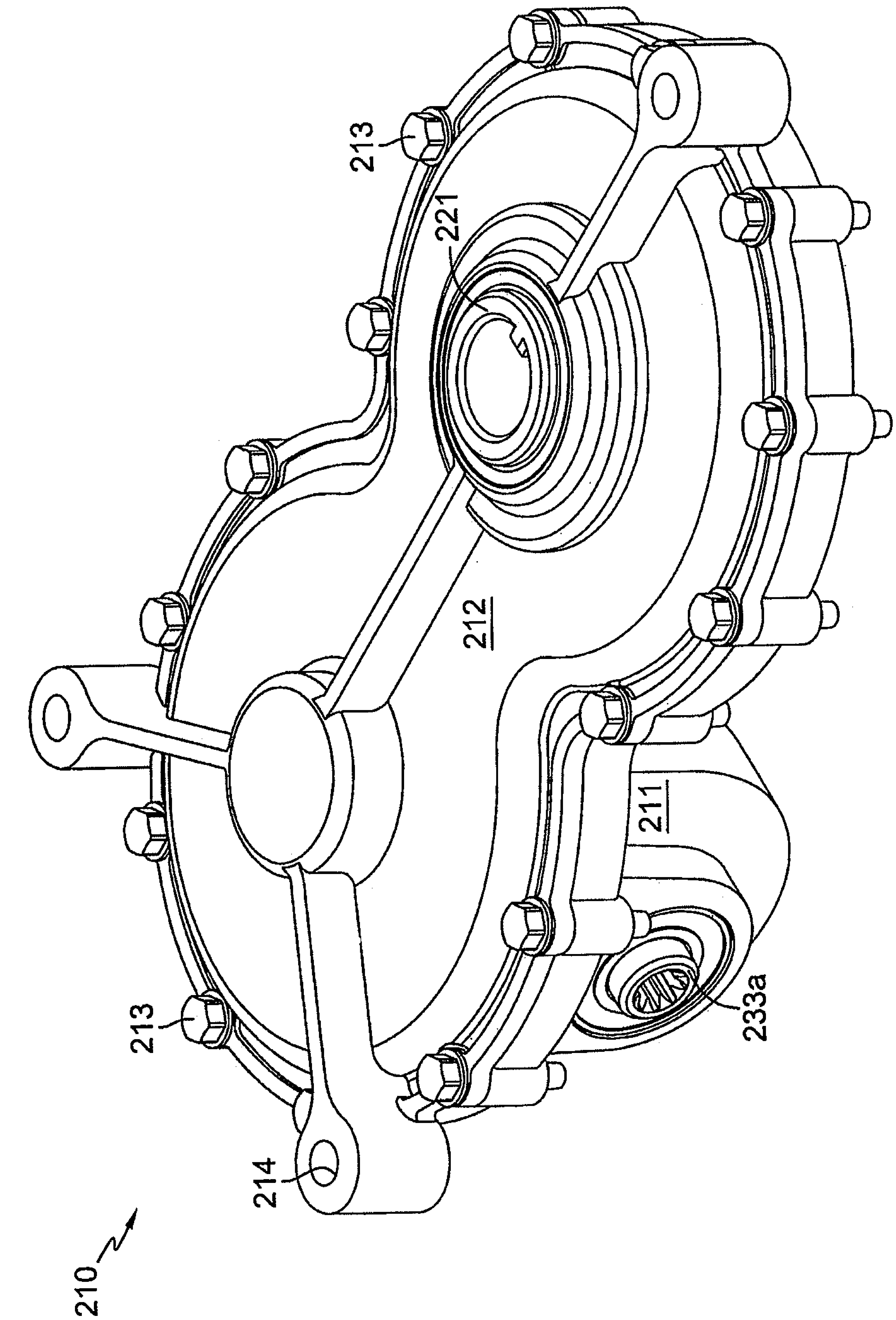

FIG. 2A

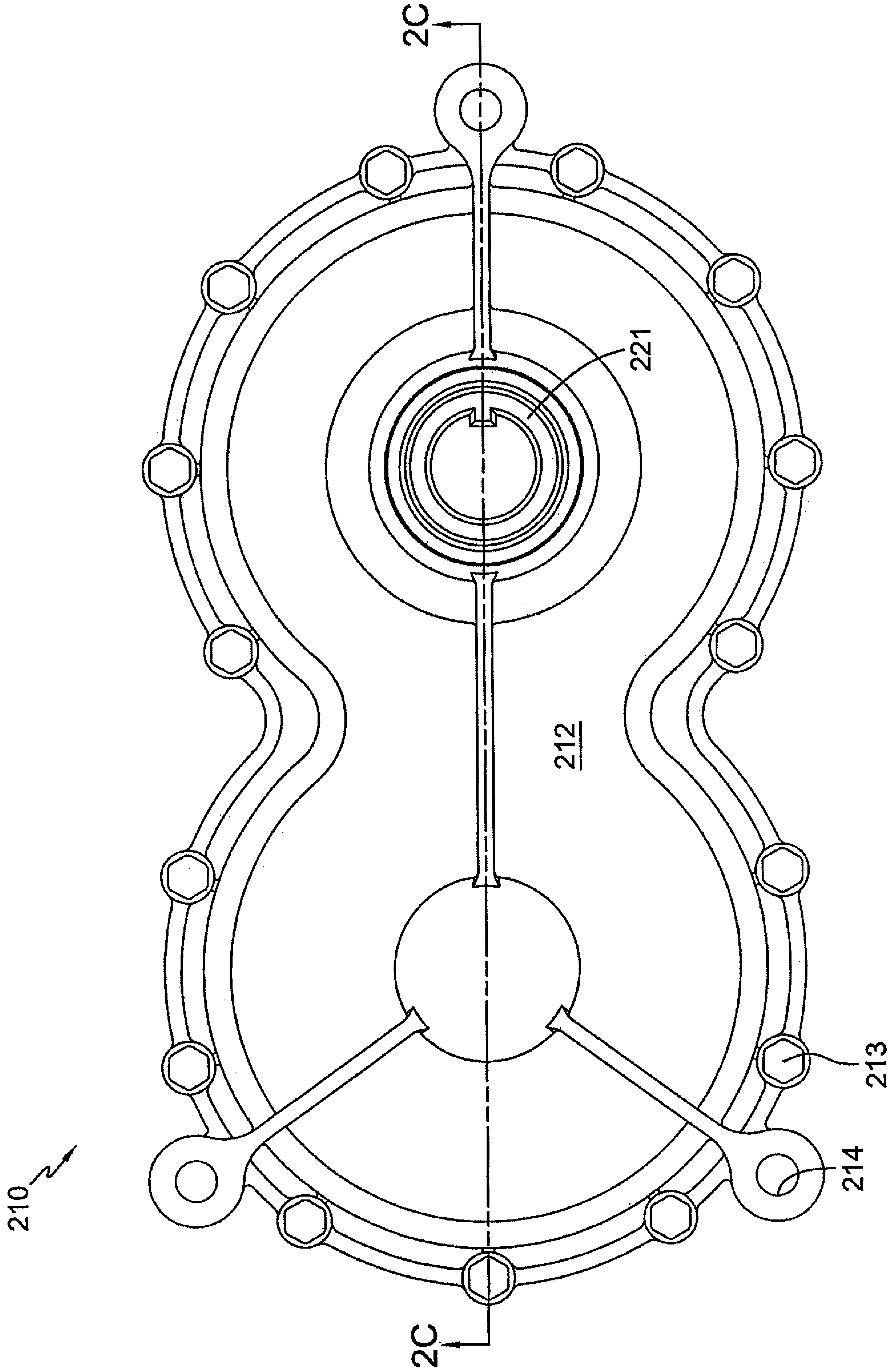

FIG. 2B

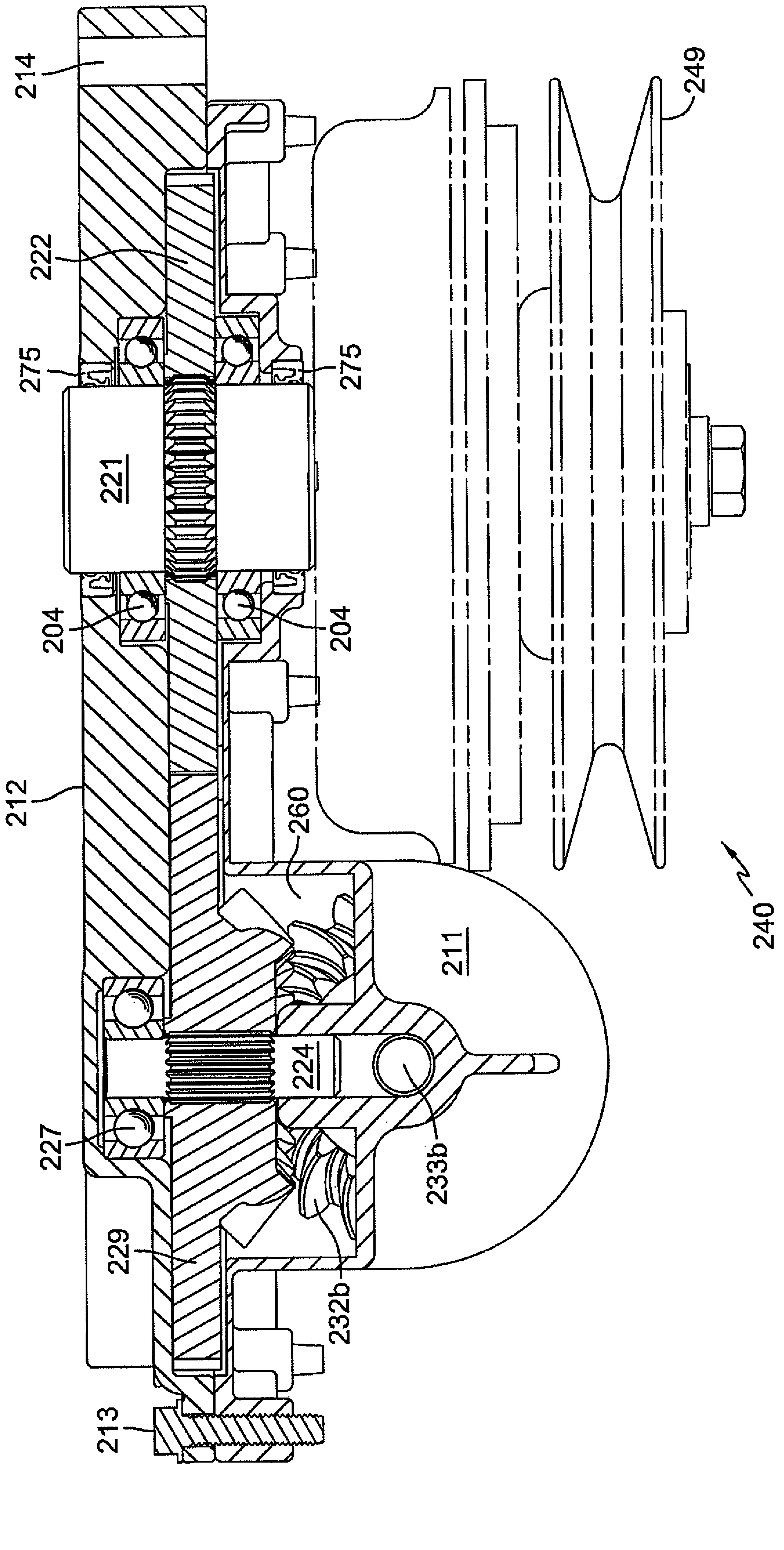
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FIG. 2C

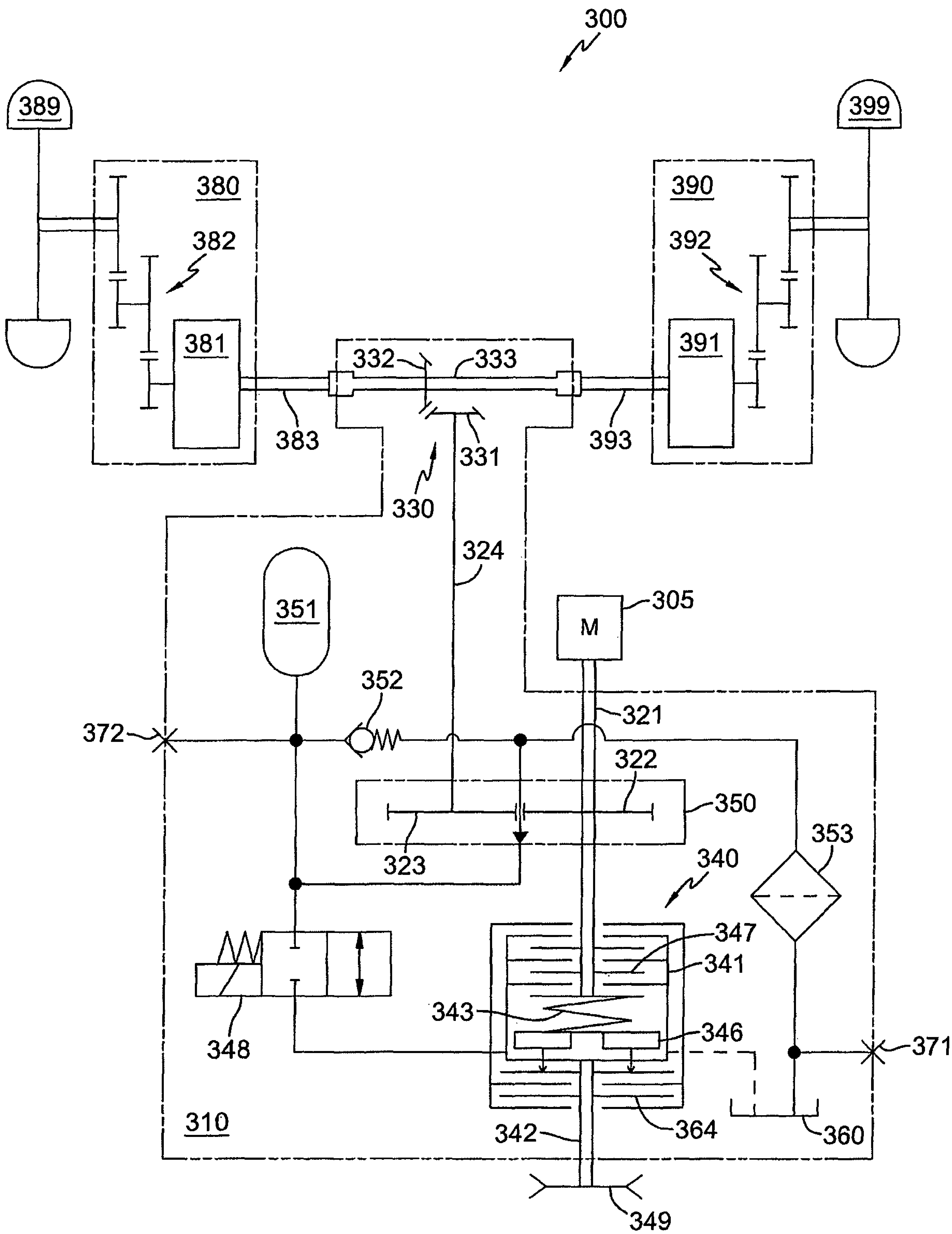
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FIG. 3

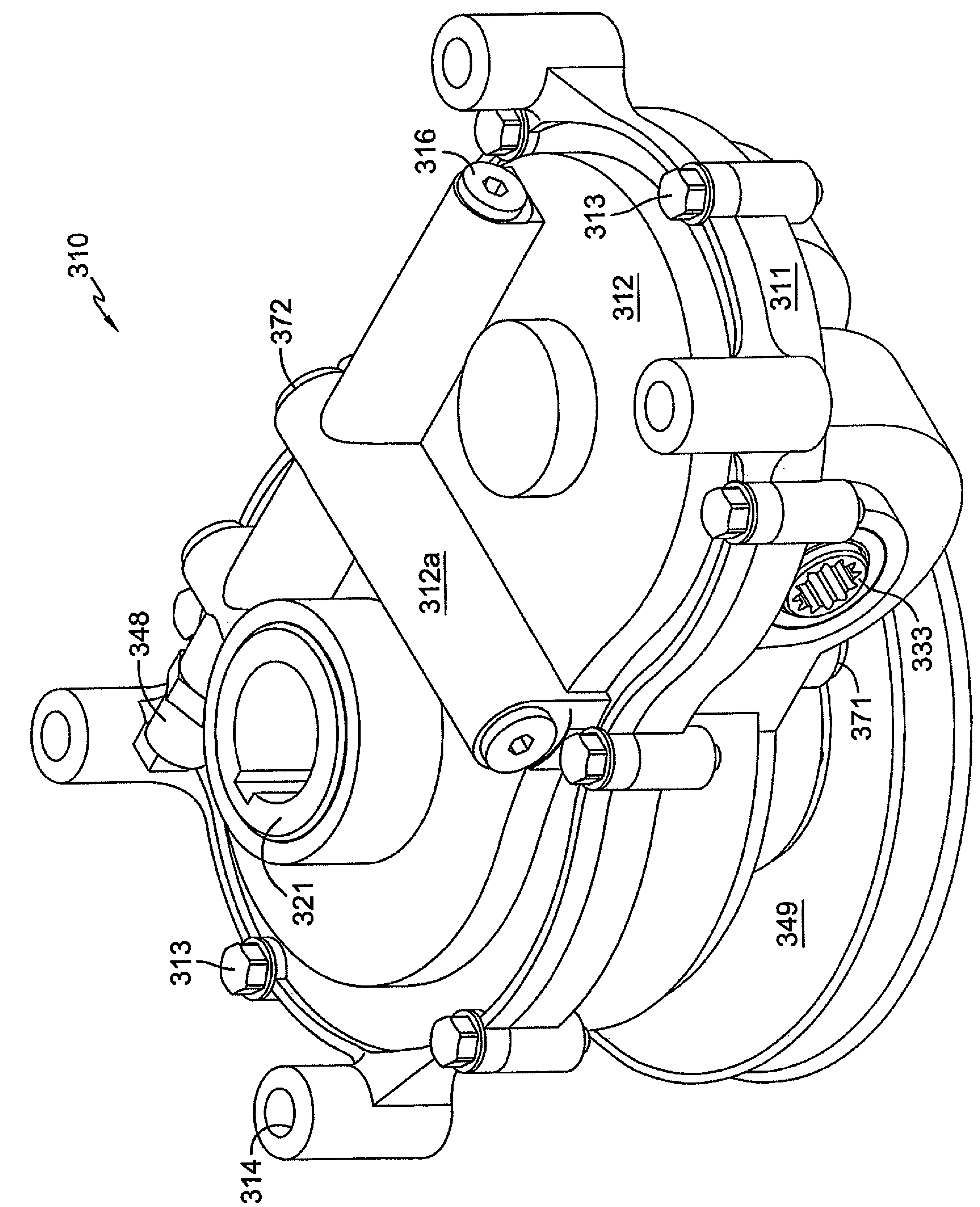
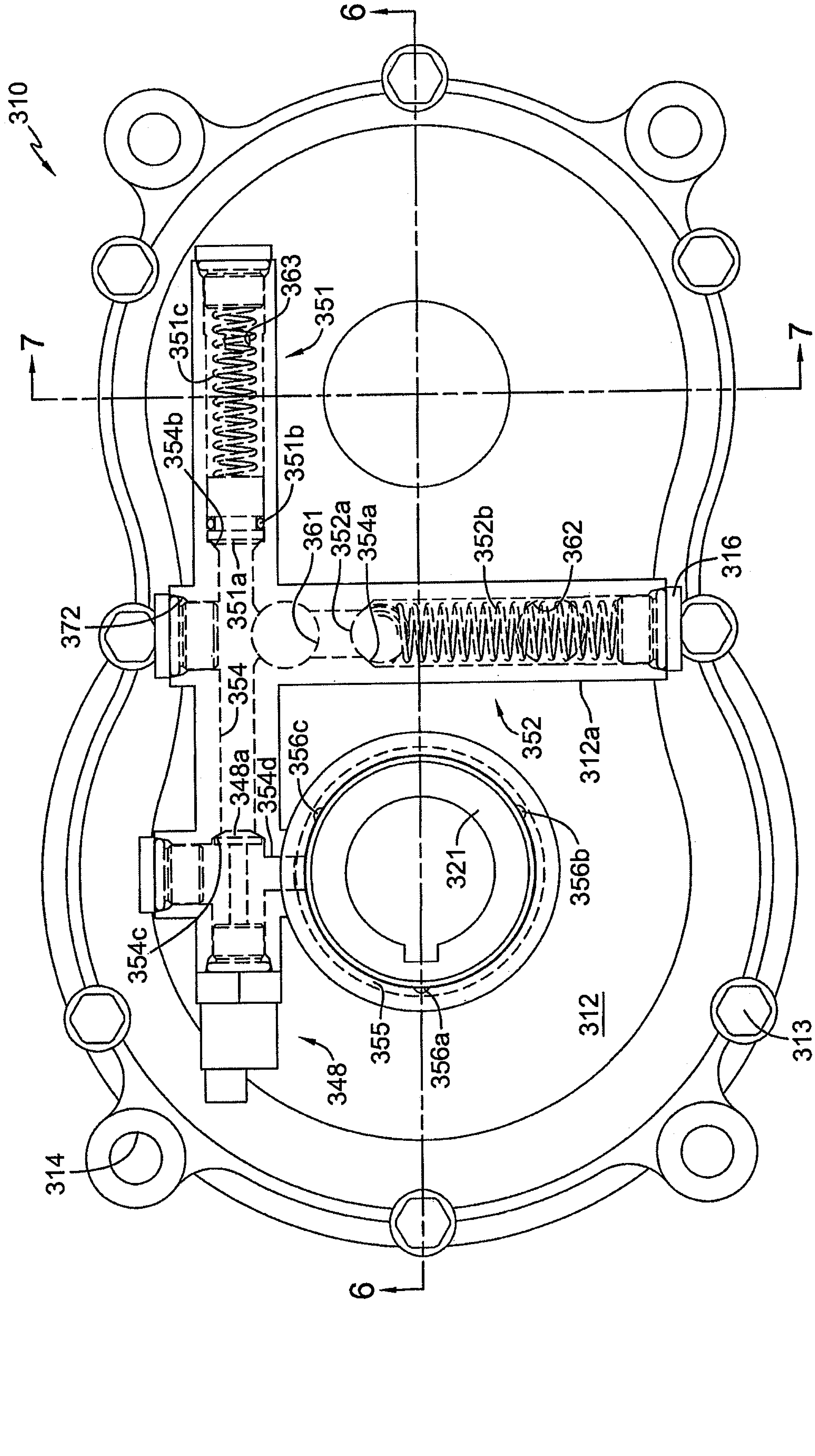


FIG. 4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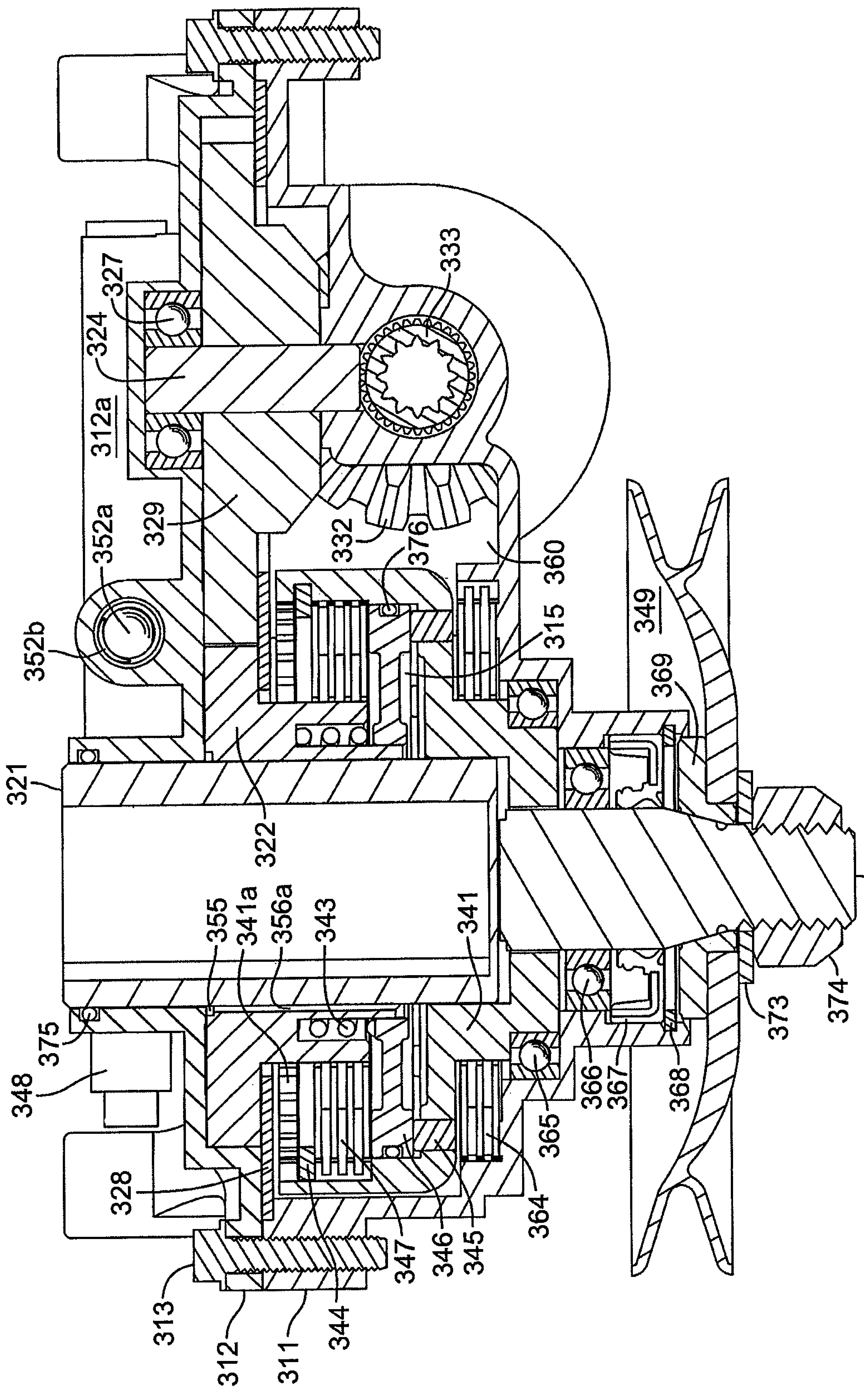
348
314

FIG. 5



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360
352b
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FIG. 6

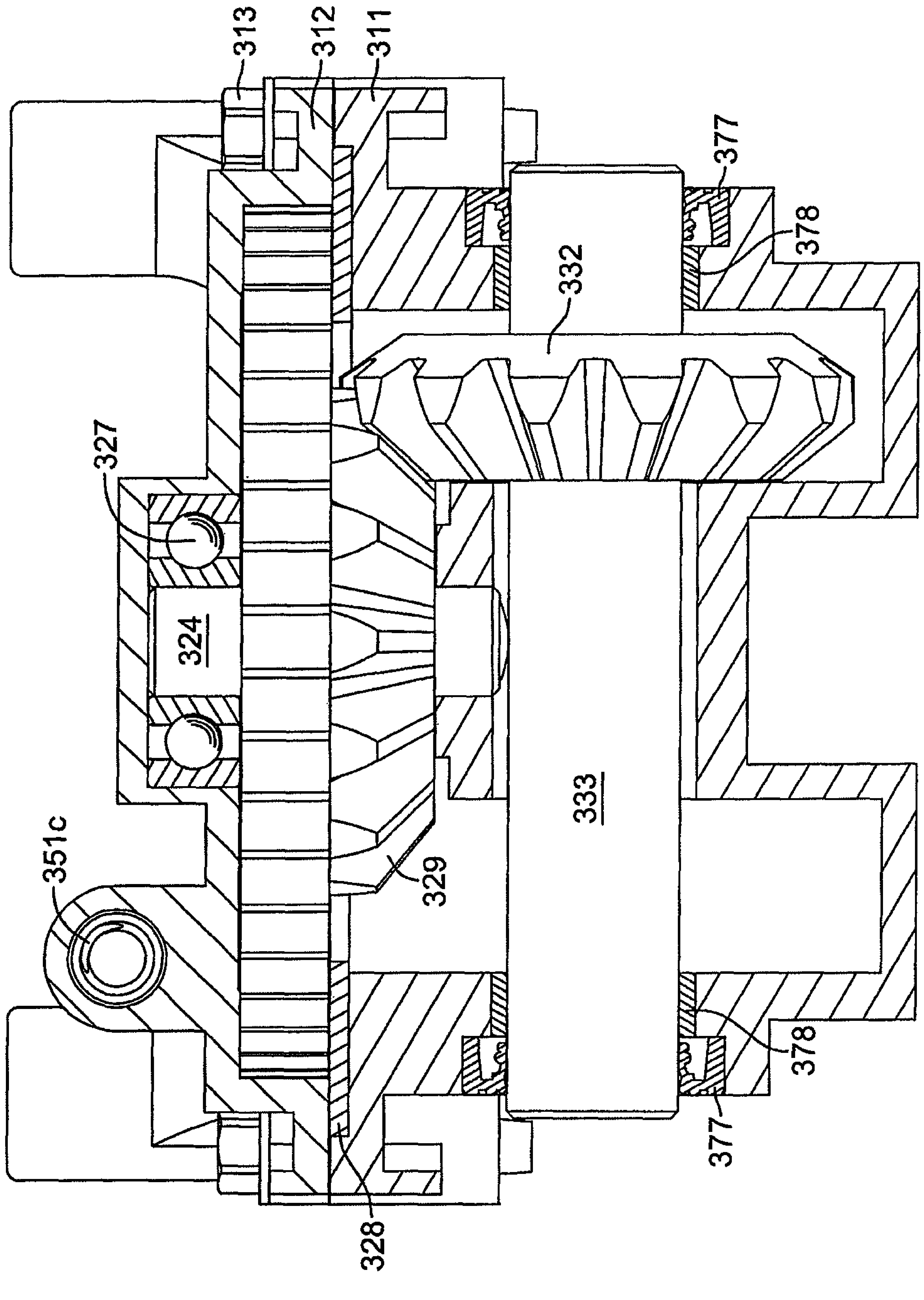

FIG. 7

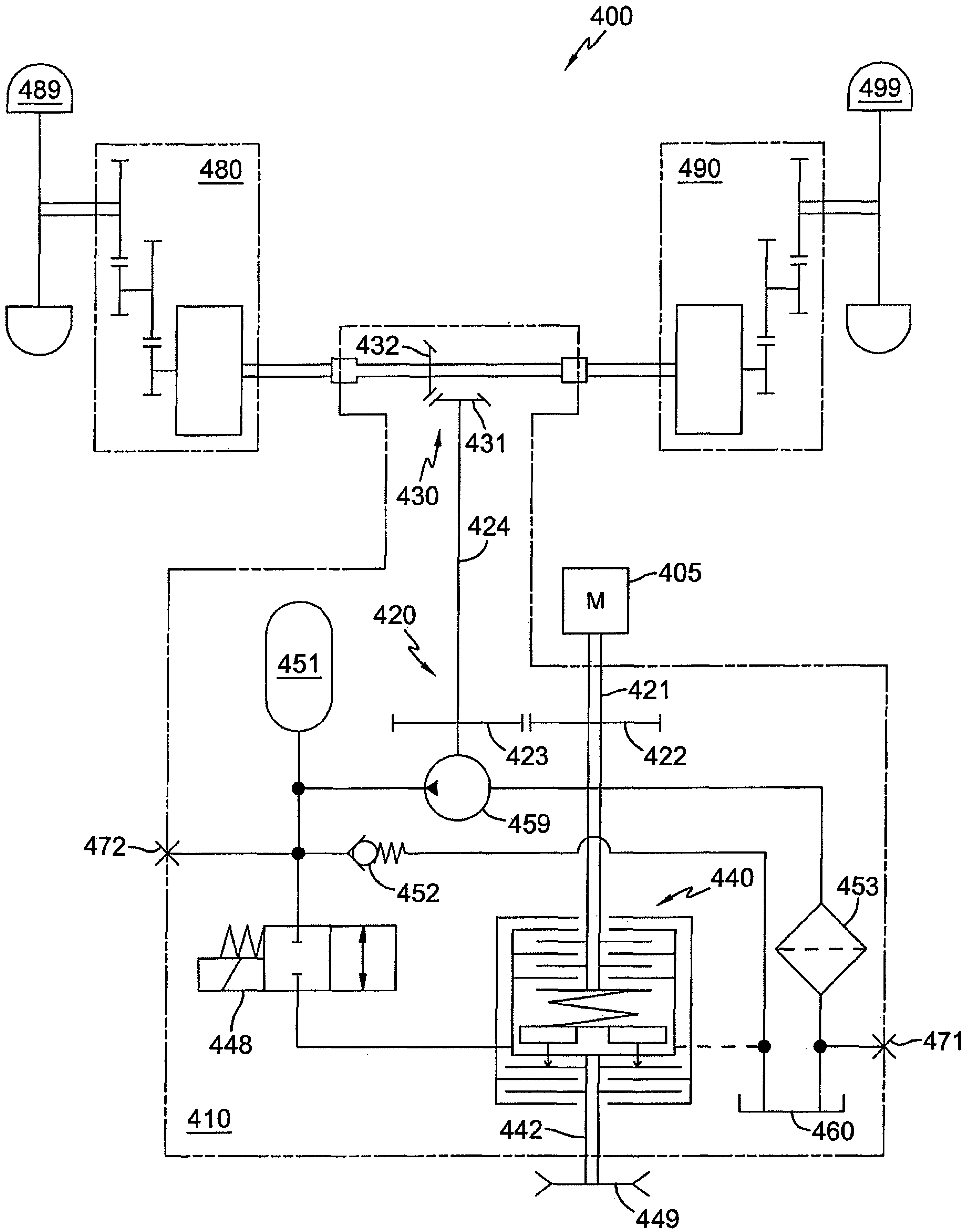
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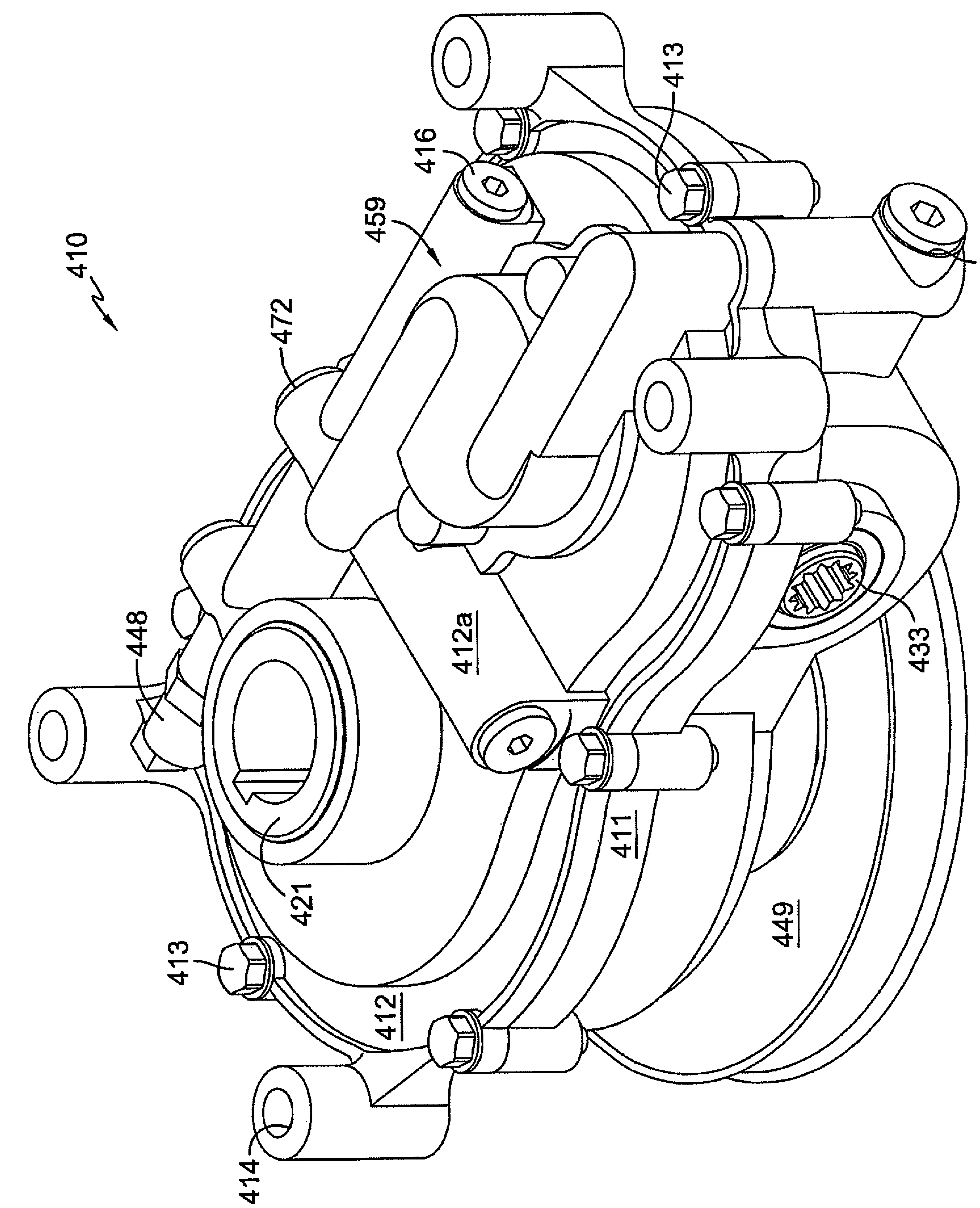
FIG. 8



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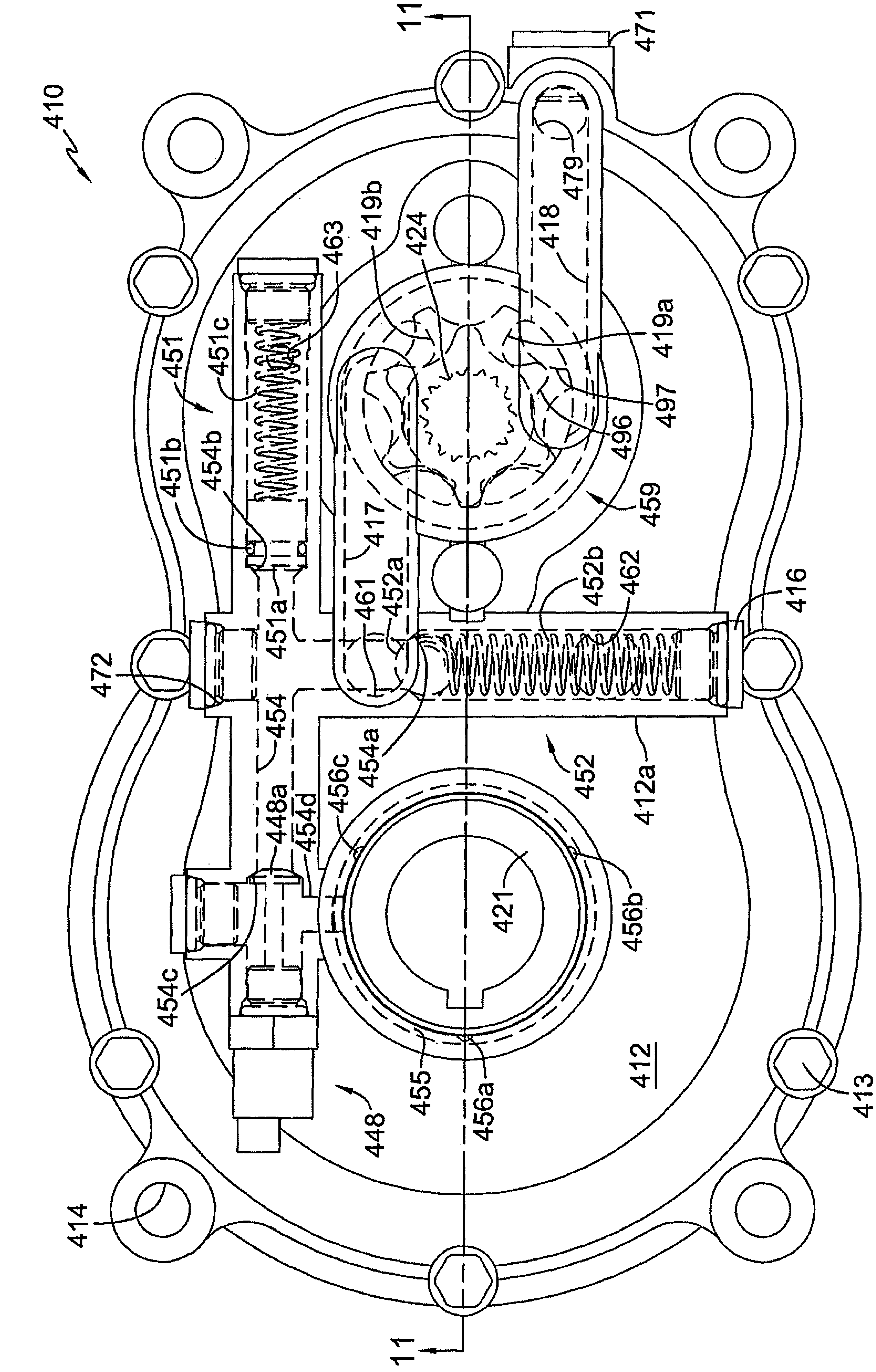

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419a
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451c
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416
461
452a
451a
472
454
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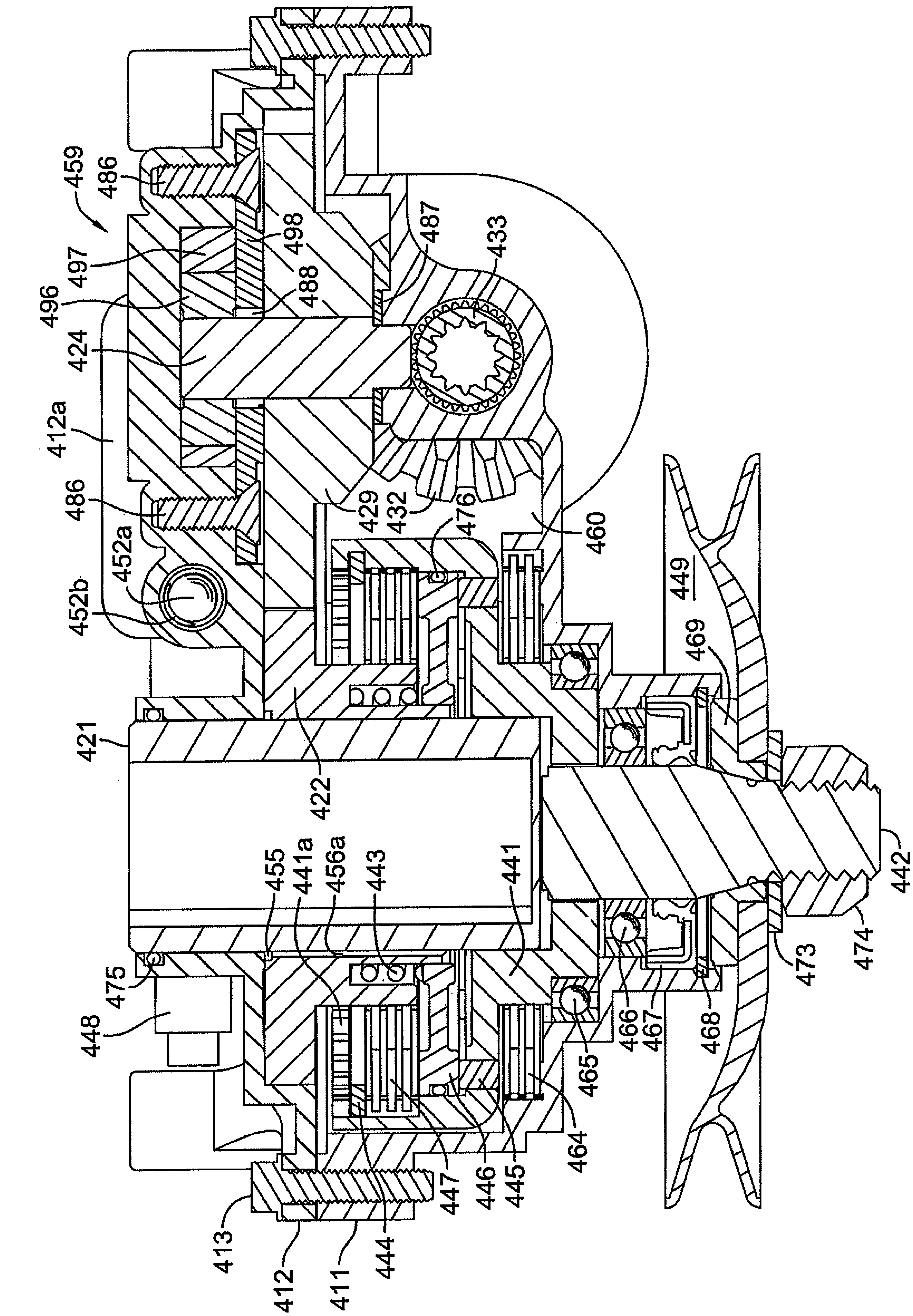
FIG. 10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441a
456a
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FIG. 11

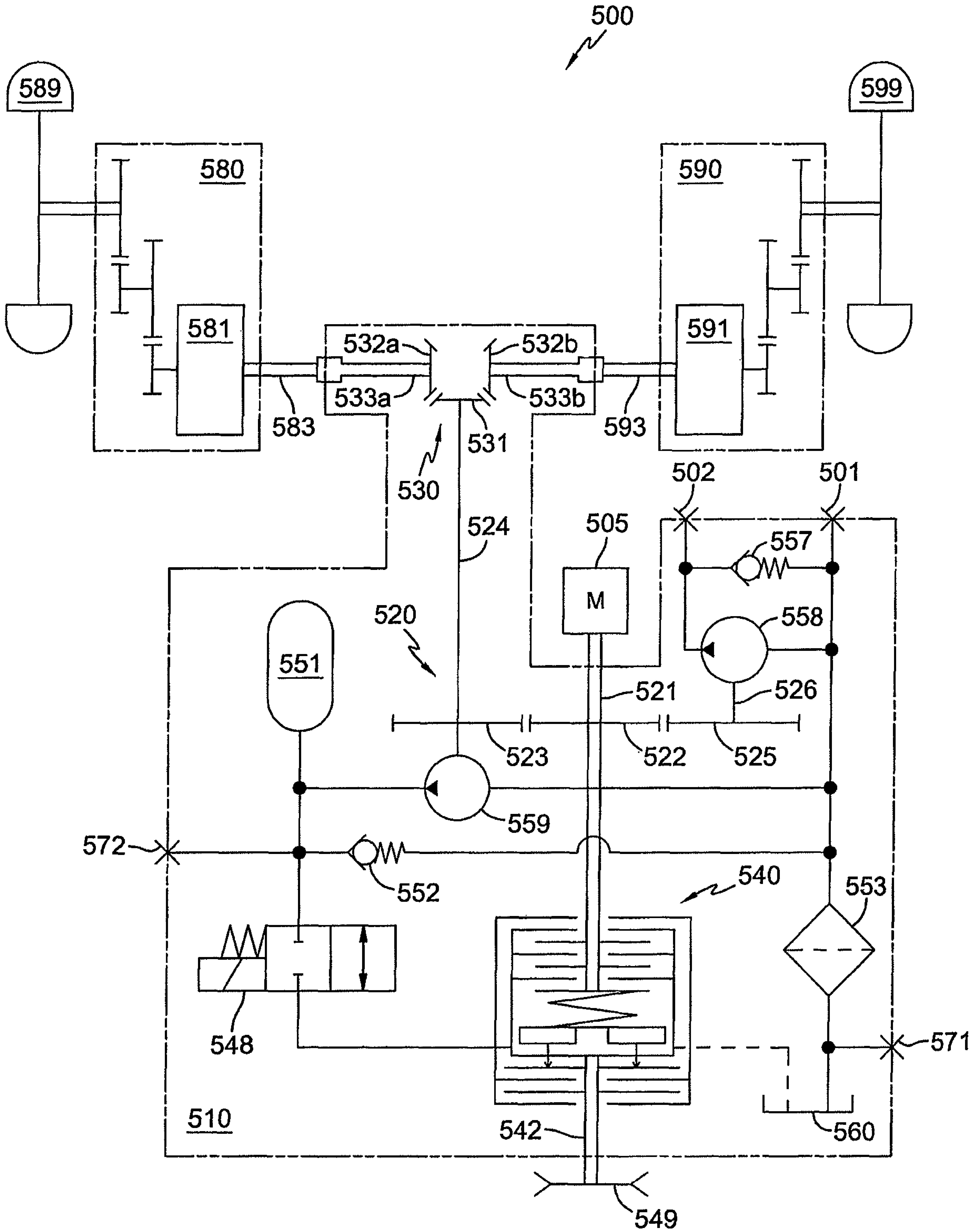
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FIG. 12

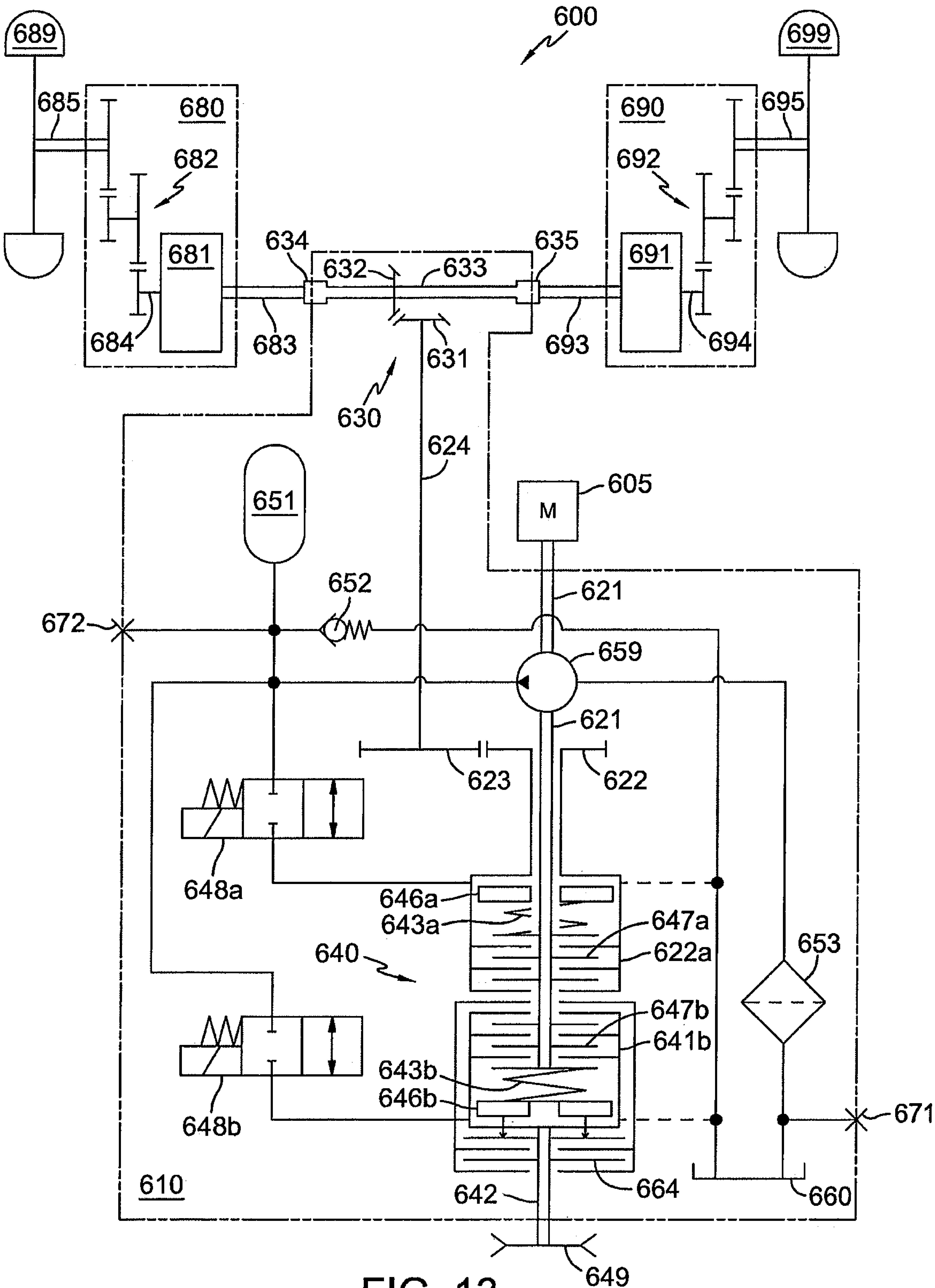
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FIG. 13

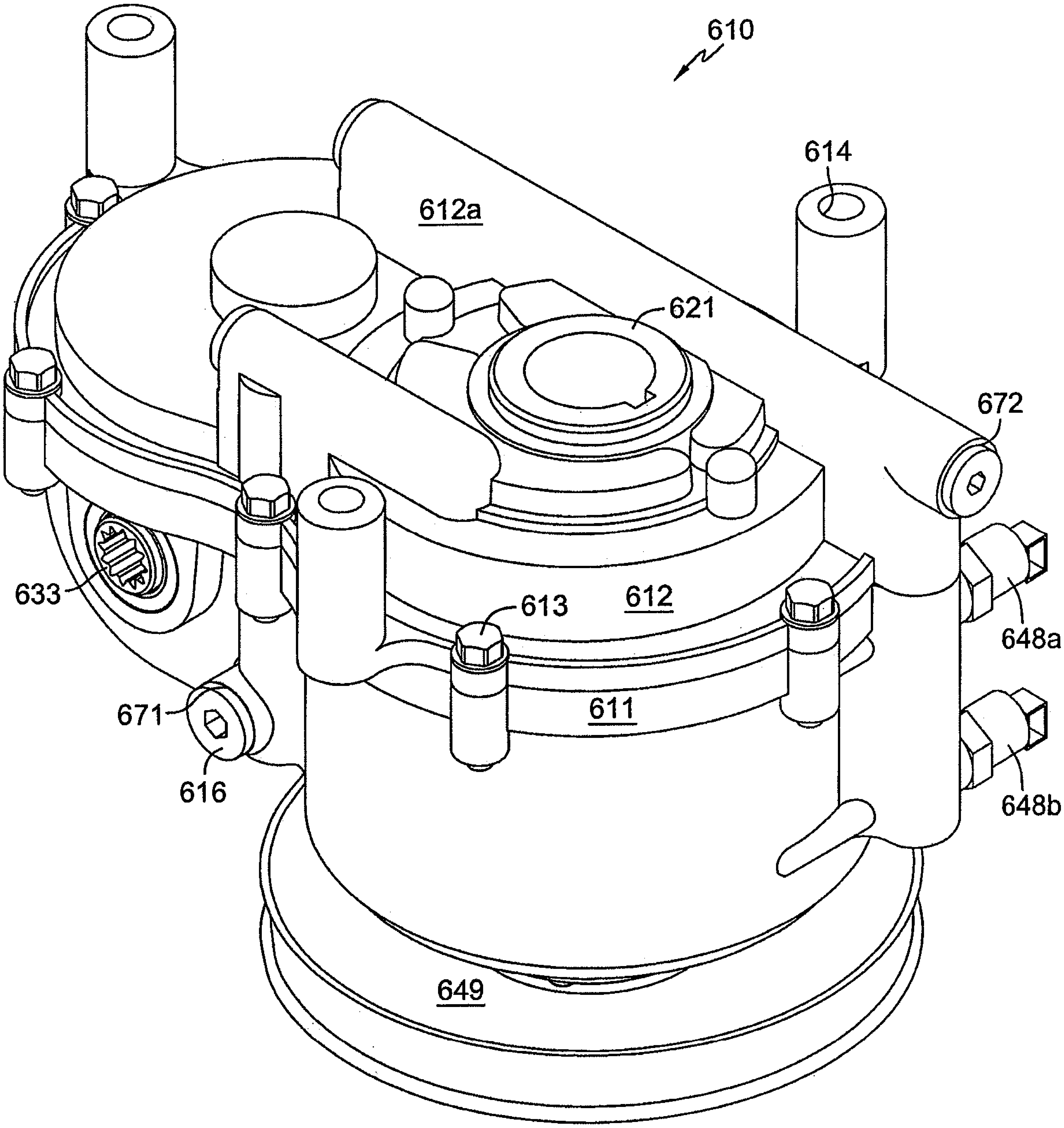

FIG. 14

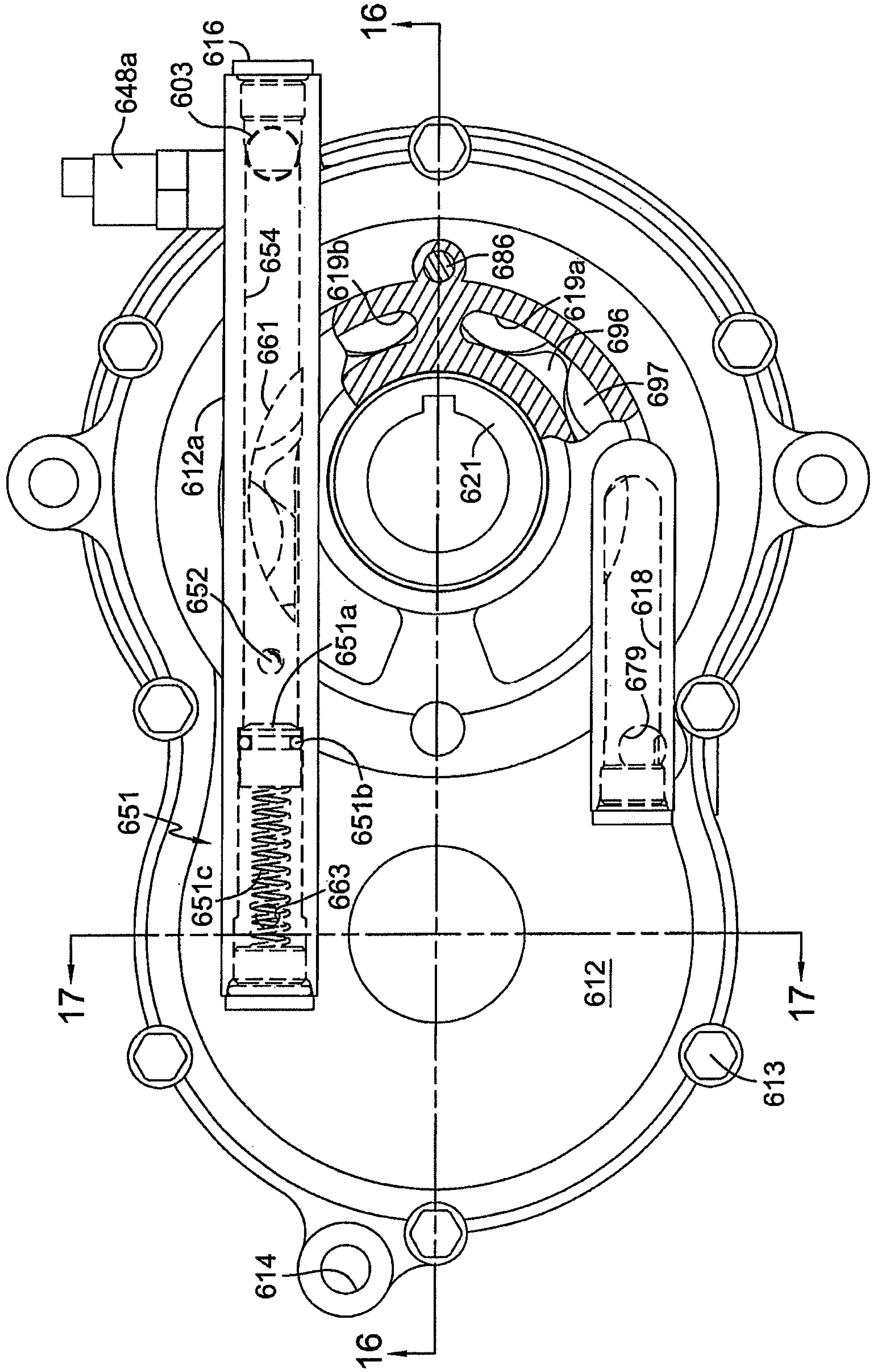
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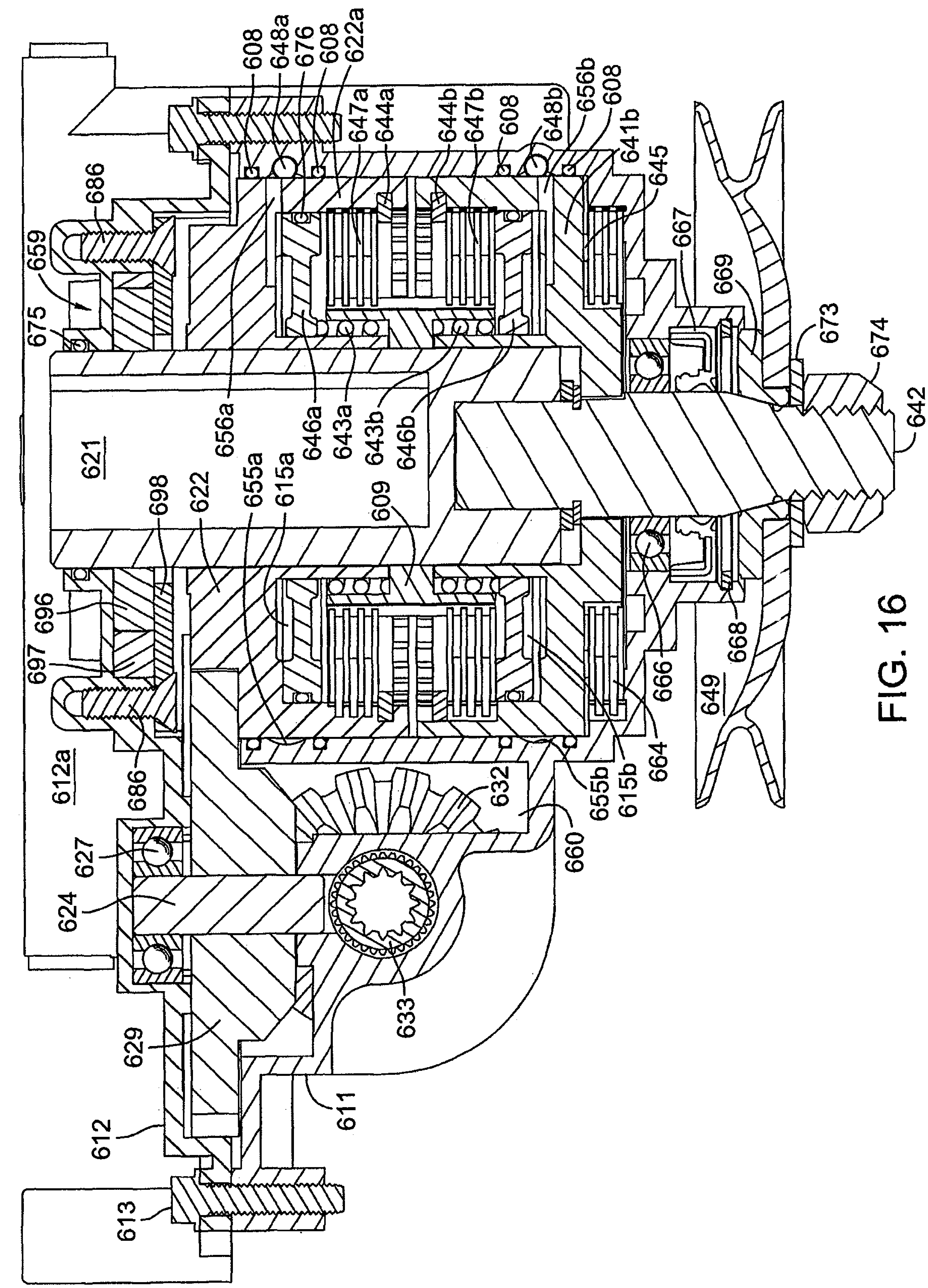
FIG. 15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667
669
673
674
642
656a
646a
643a
643b
646b
621
698
622
655a
615a
609
696
697
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FIG. 16

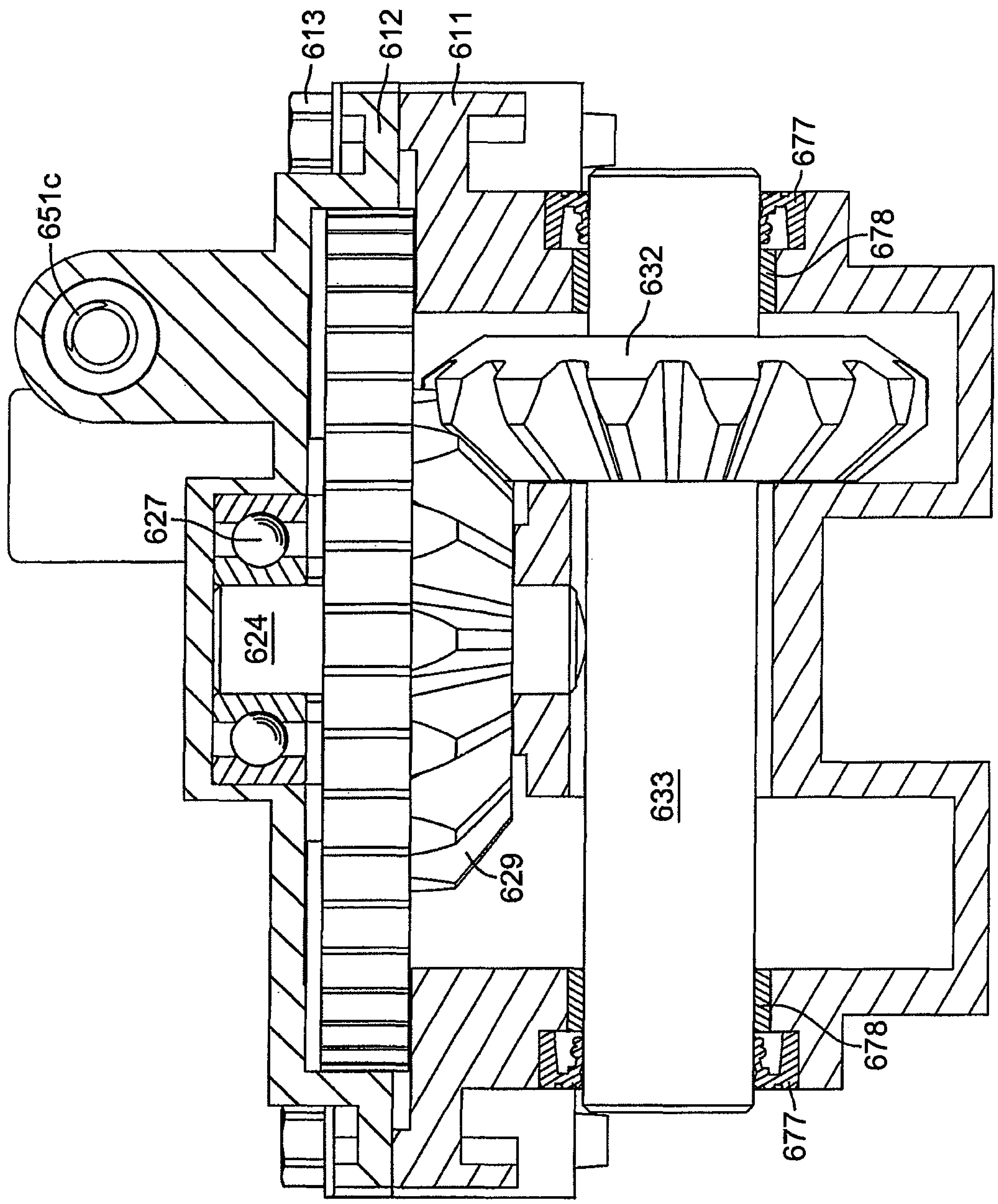
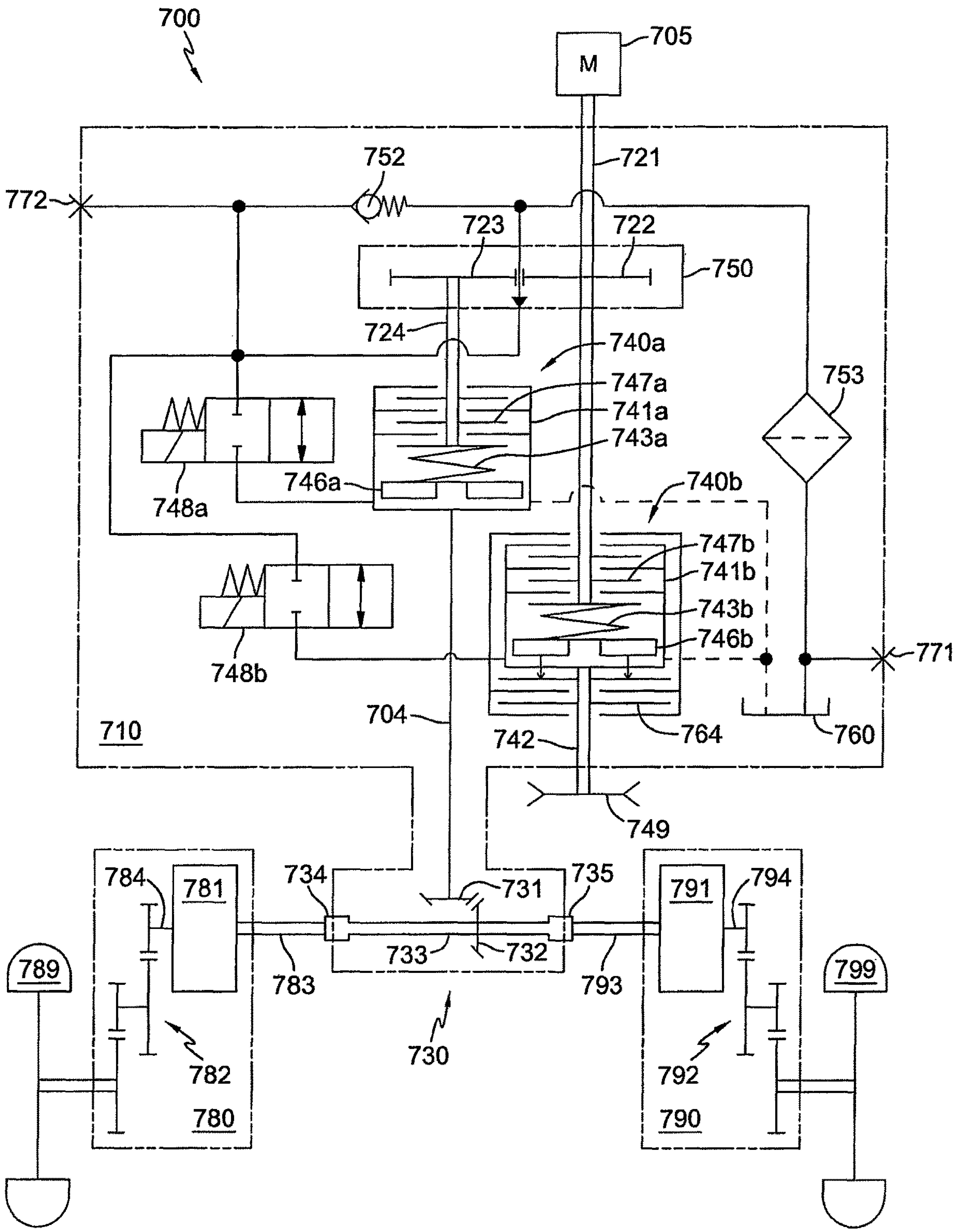


FIG. 17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784
781
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794
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733
732
799
783
793
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790

FIG. 18

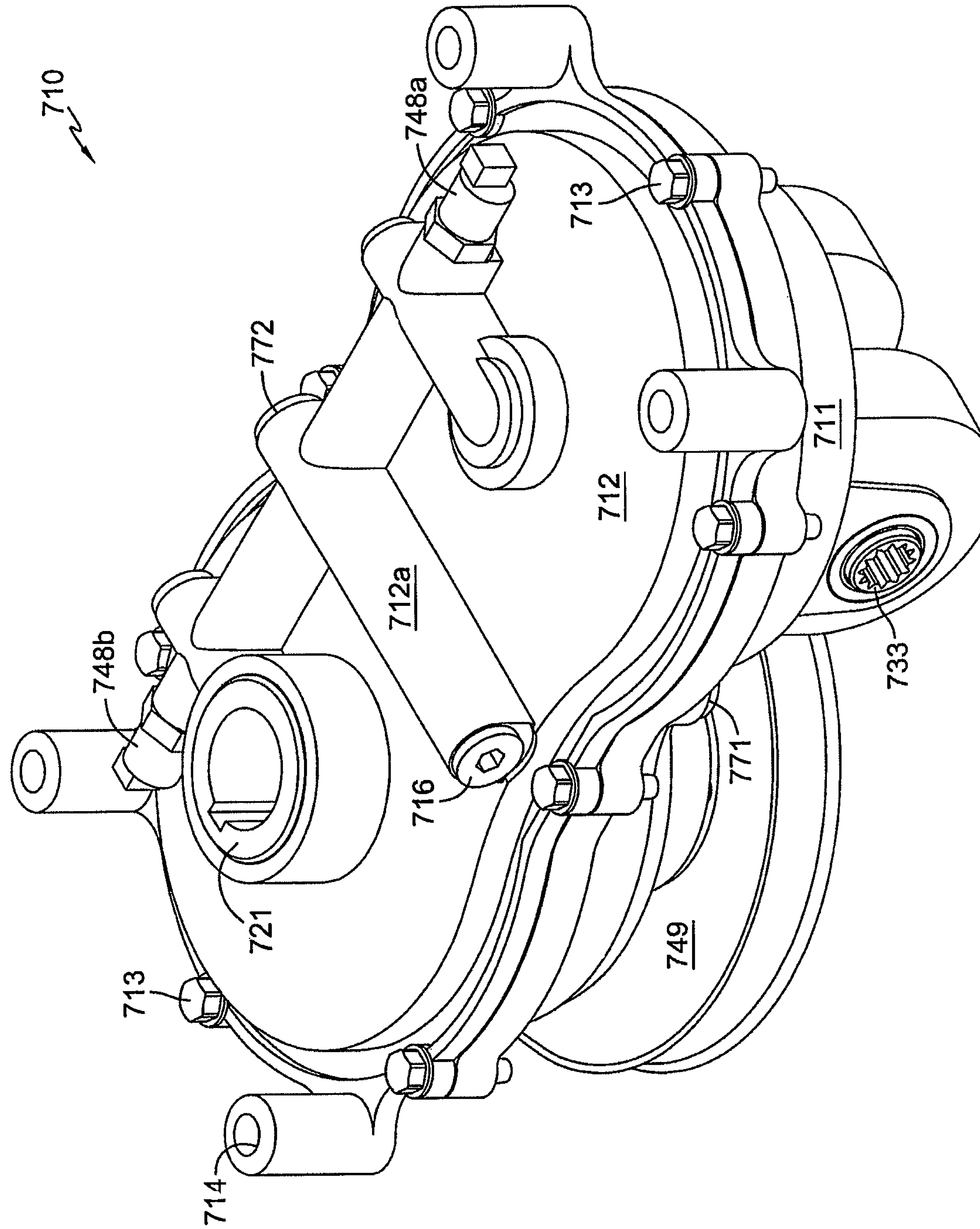
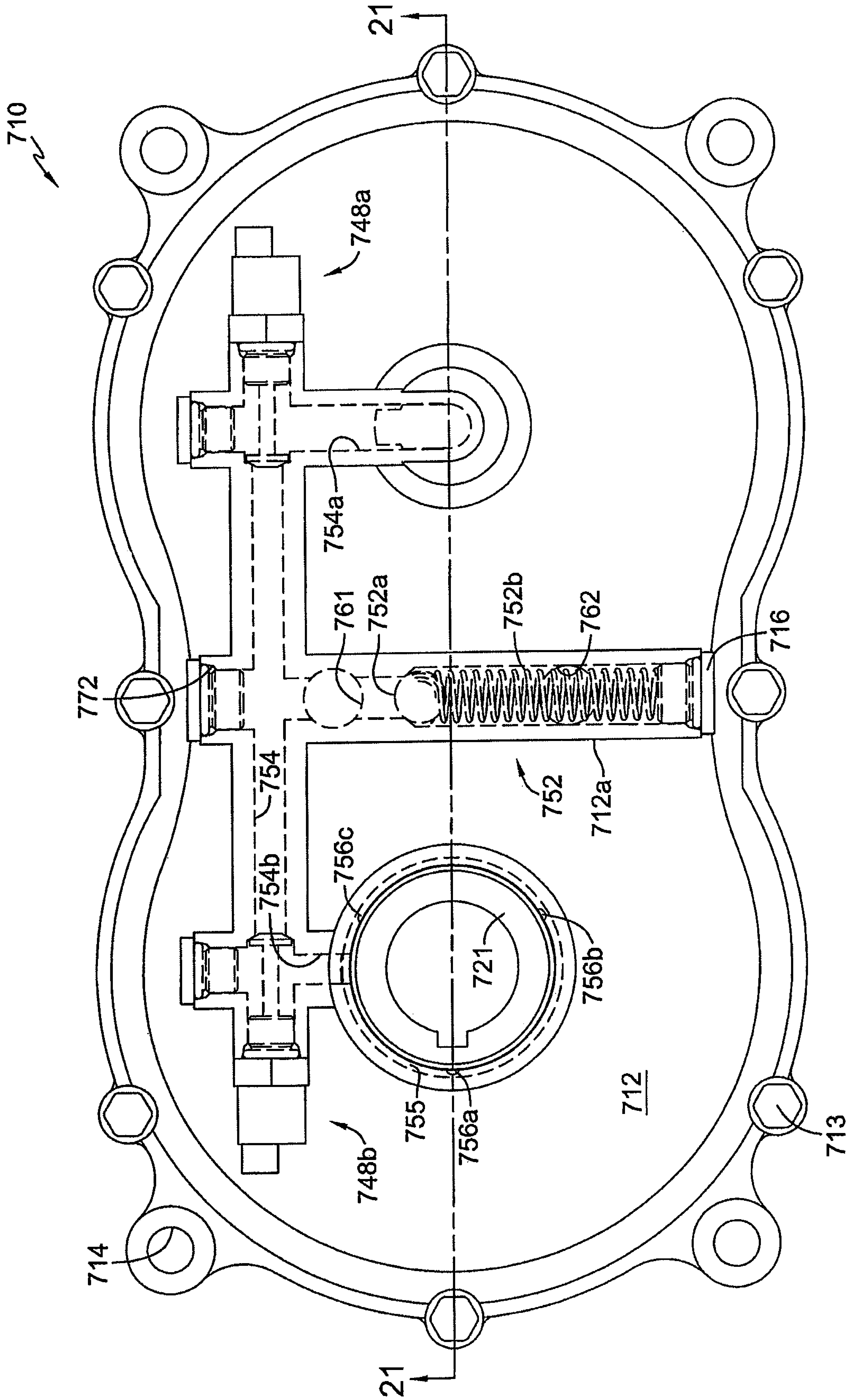


FIG. 19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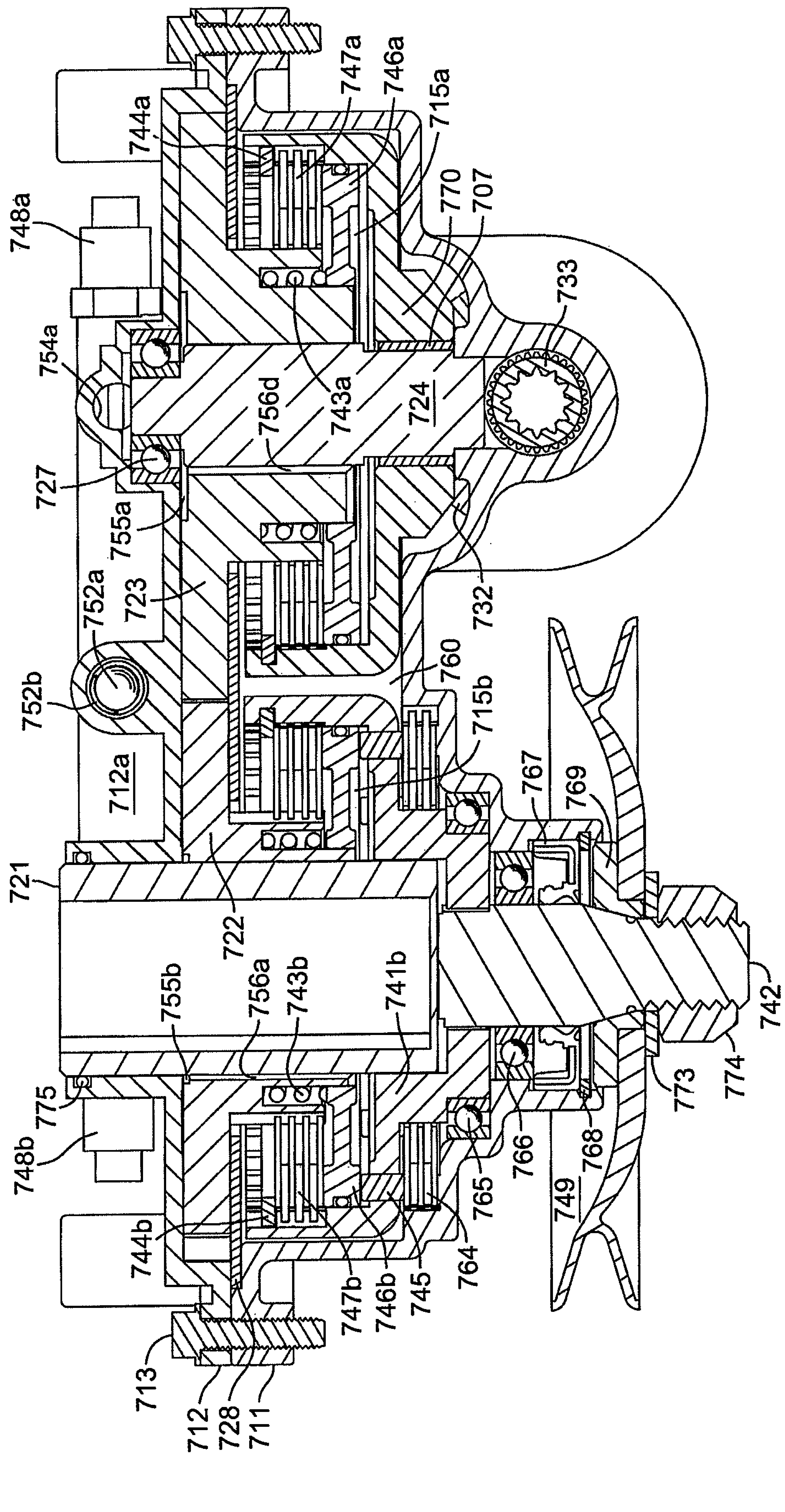
756a
712
713
748b
714
21

FIG. 20



748a
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756d
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732
752b
760
715b
712a
767
769
721
722
755b
756a
743b
741b
742
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712
728
711

FIG. 21

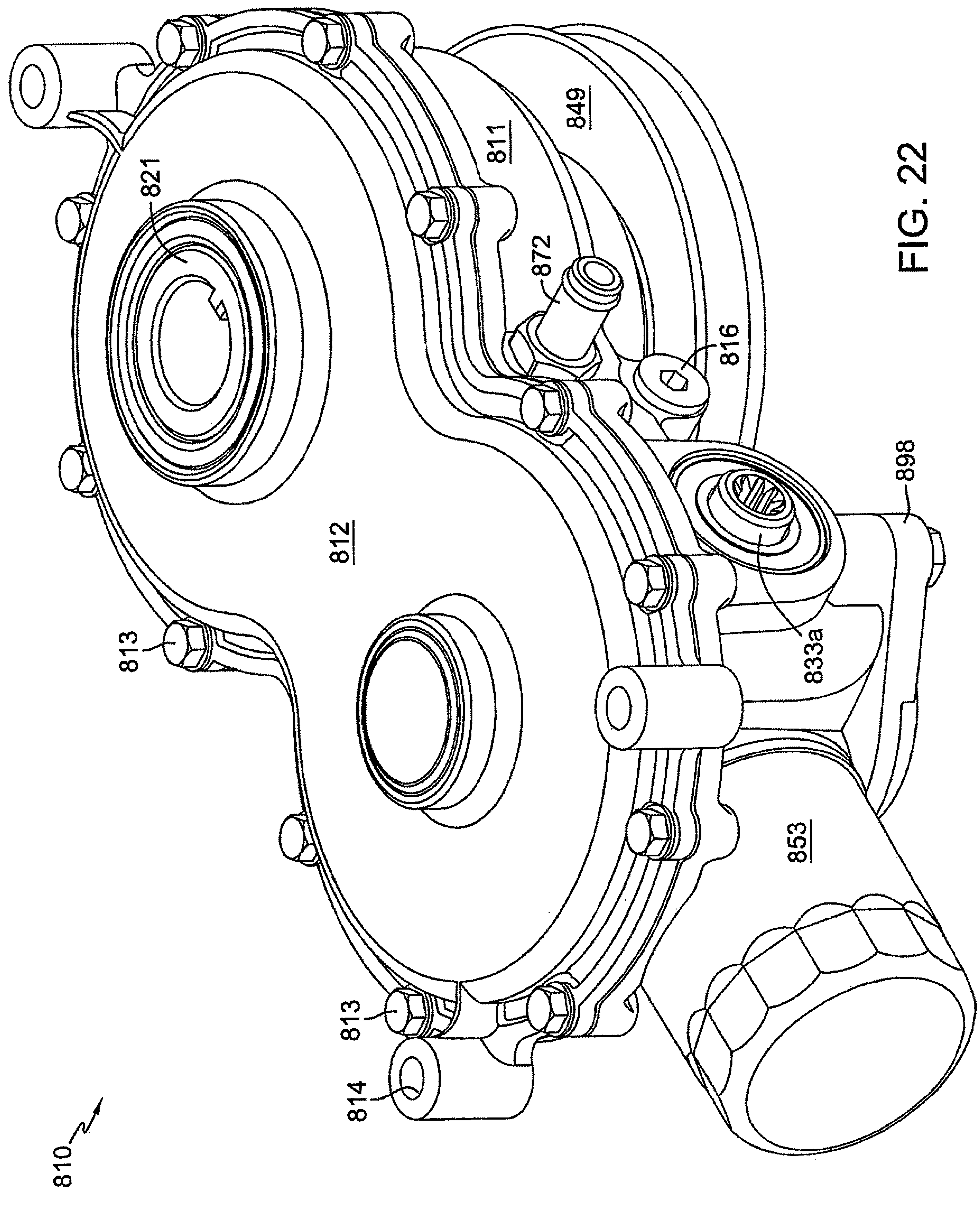


FIG. 22

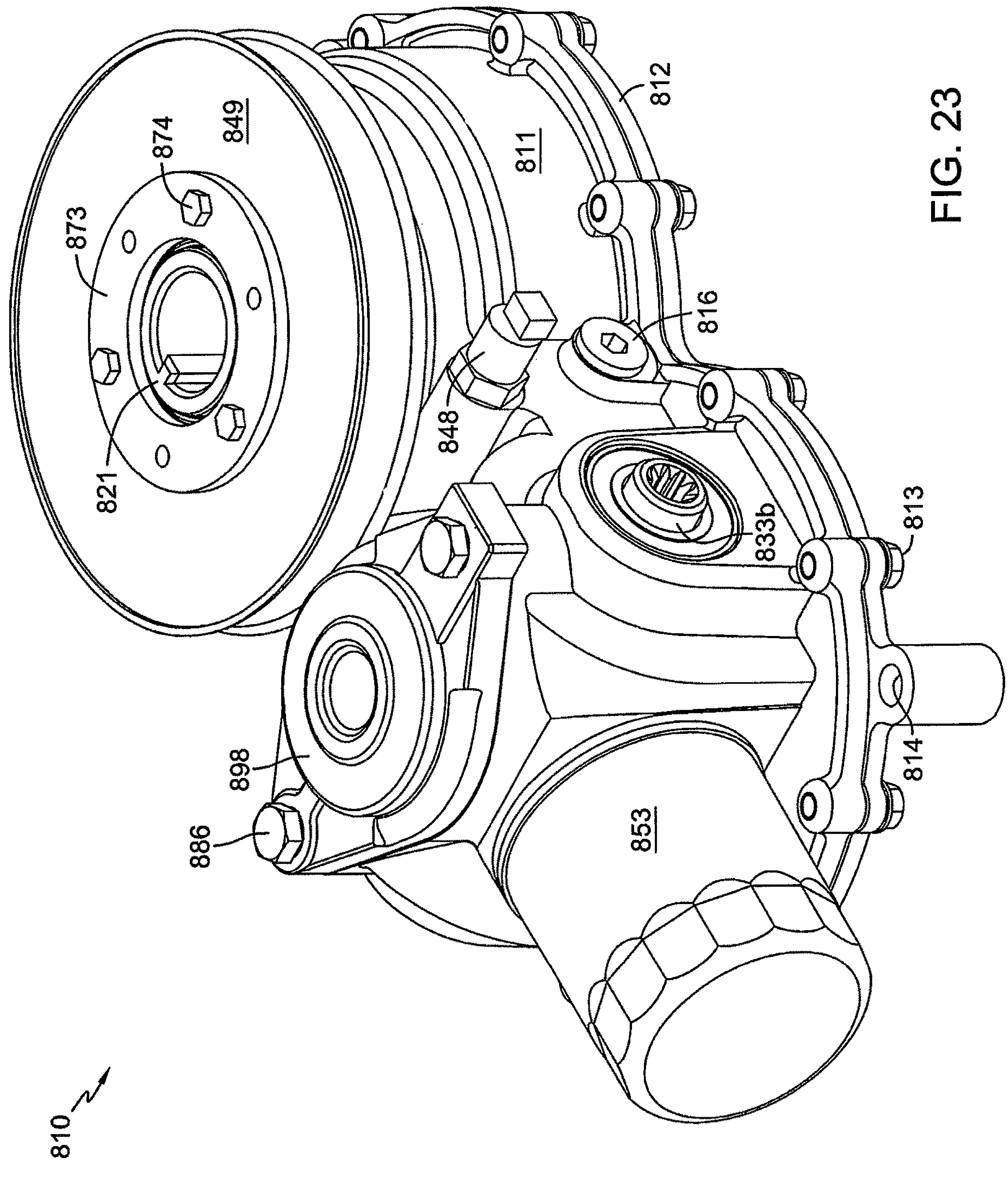
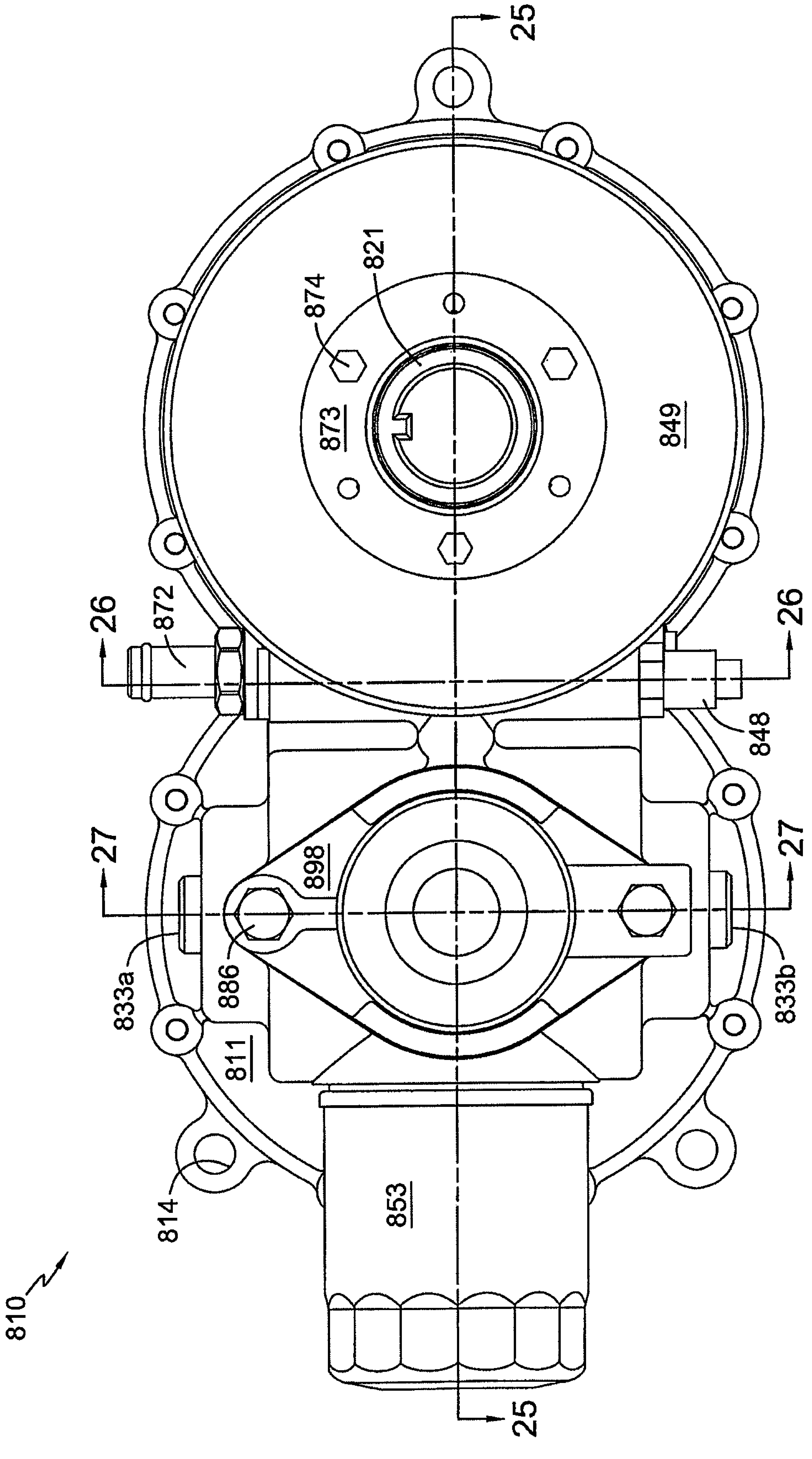


FIG. 23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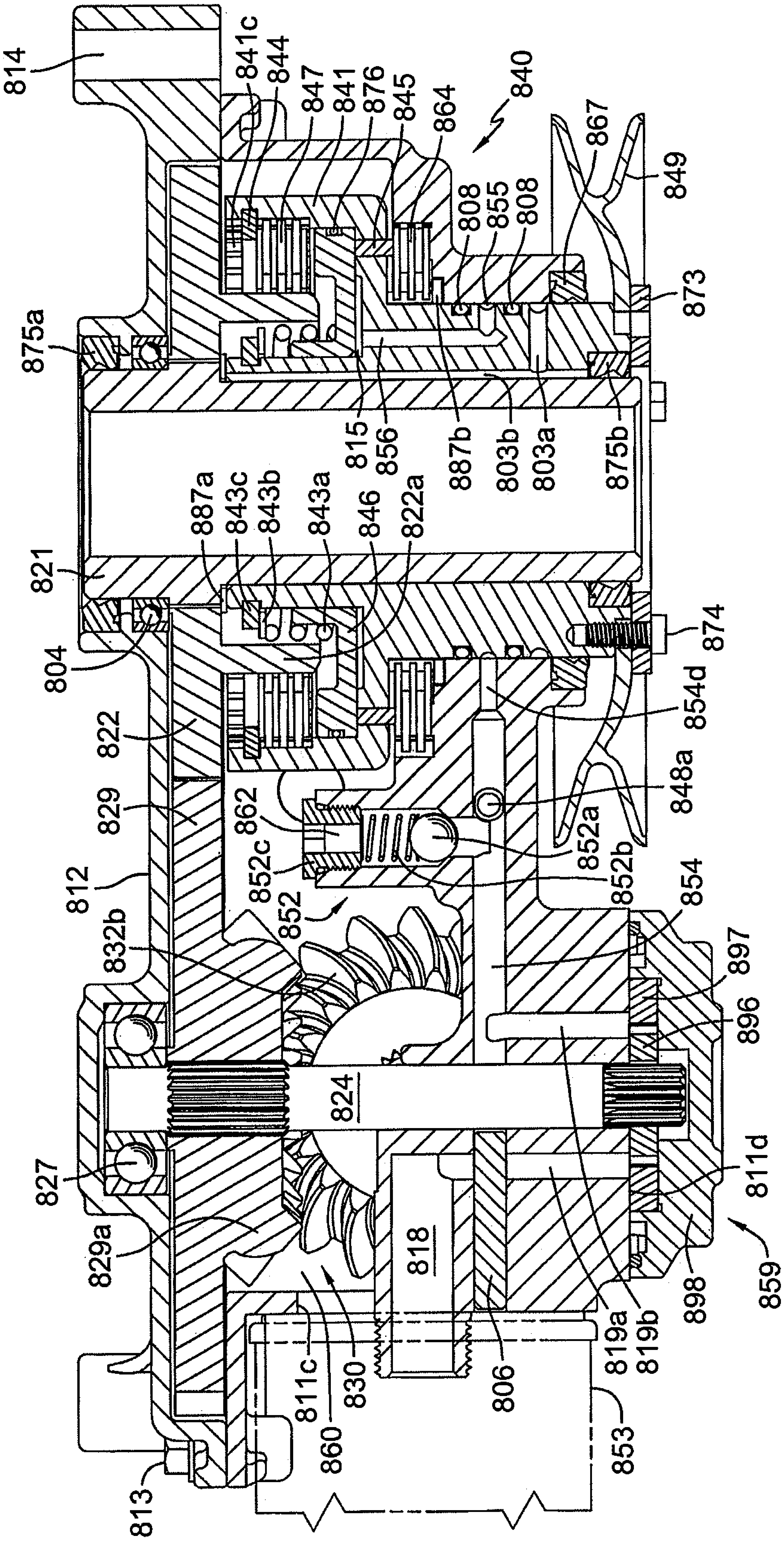
810
25

FIG. 24



814
841c
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808
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803b
803a
875b
821
887a
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843b
843a
846
822a
874
804
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854d
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852b
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832b
897
896
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898
819a
819b
811c
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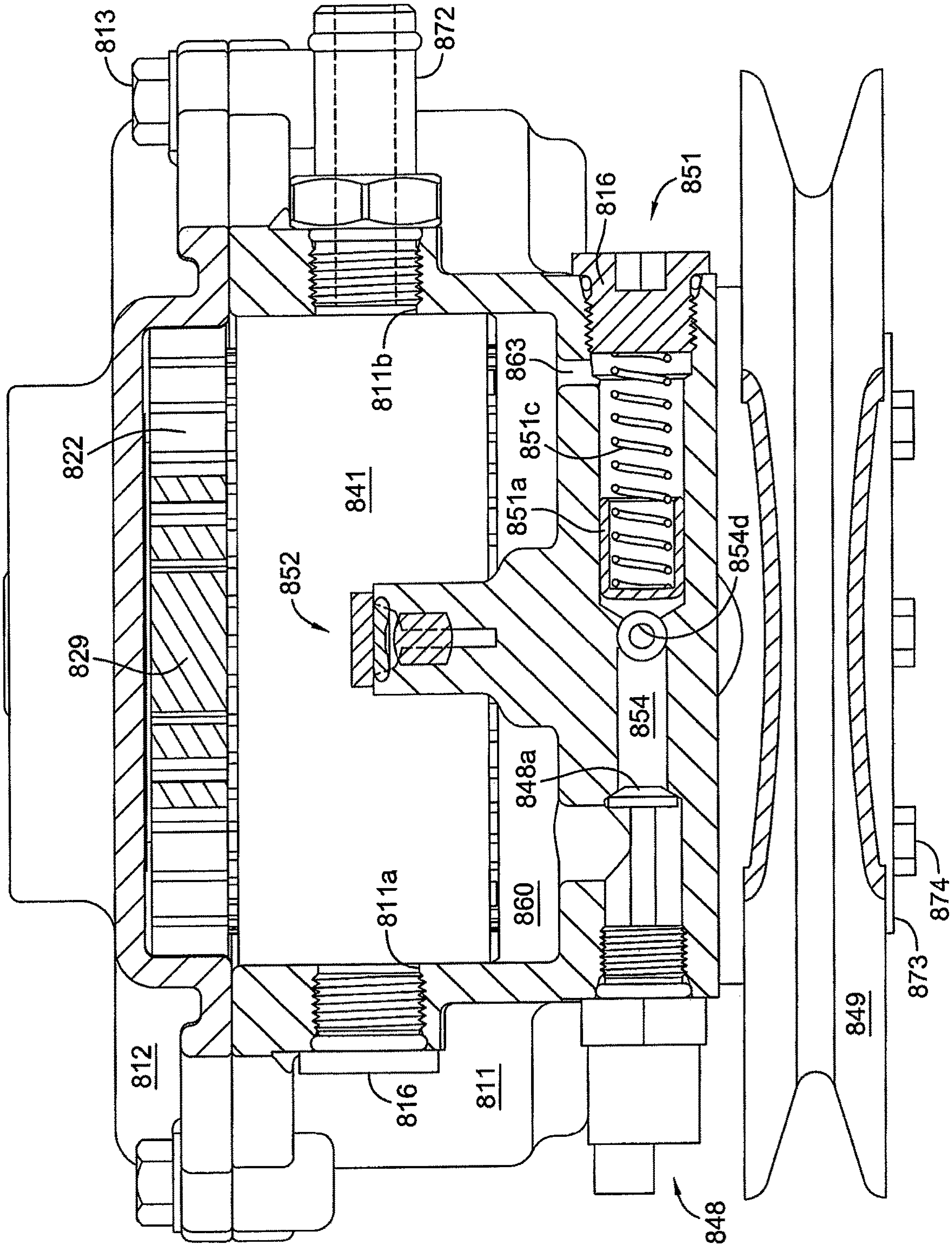
860
853
813

FIG. 25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874
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849
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816
811
848

FIG. 26

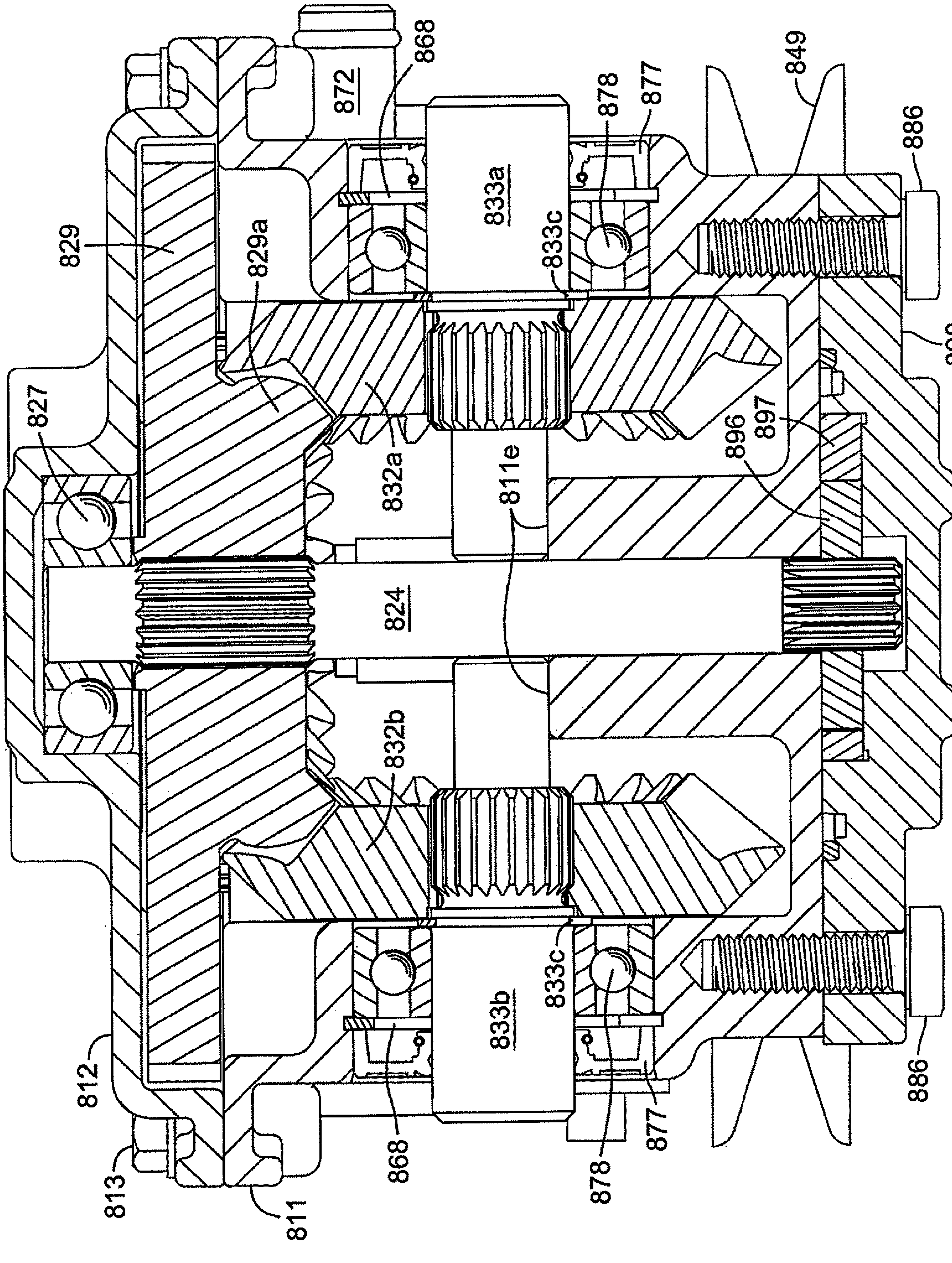

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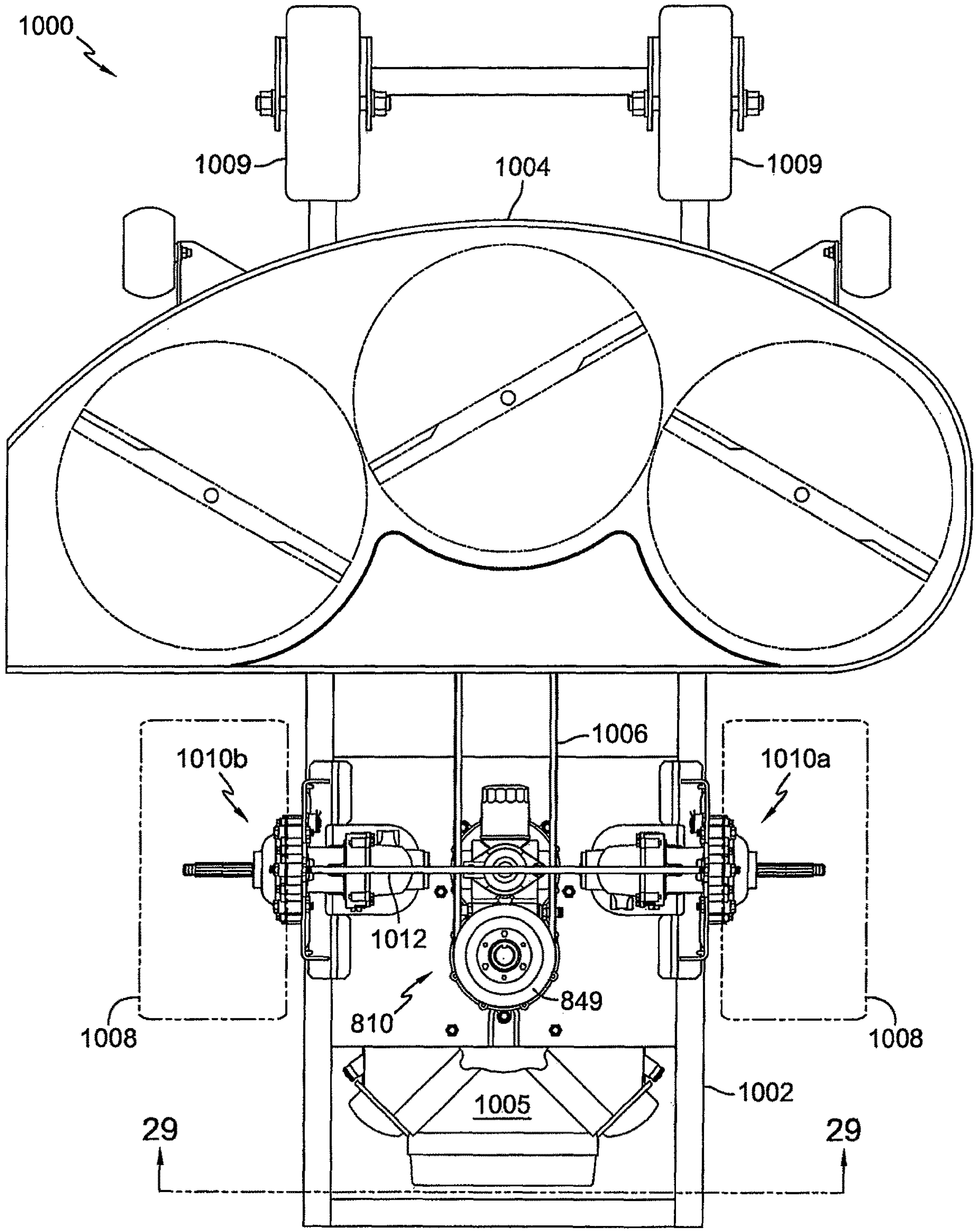
FIG. 27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849
810
1008
1008
1002
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29
29

FIG. 28

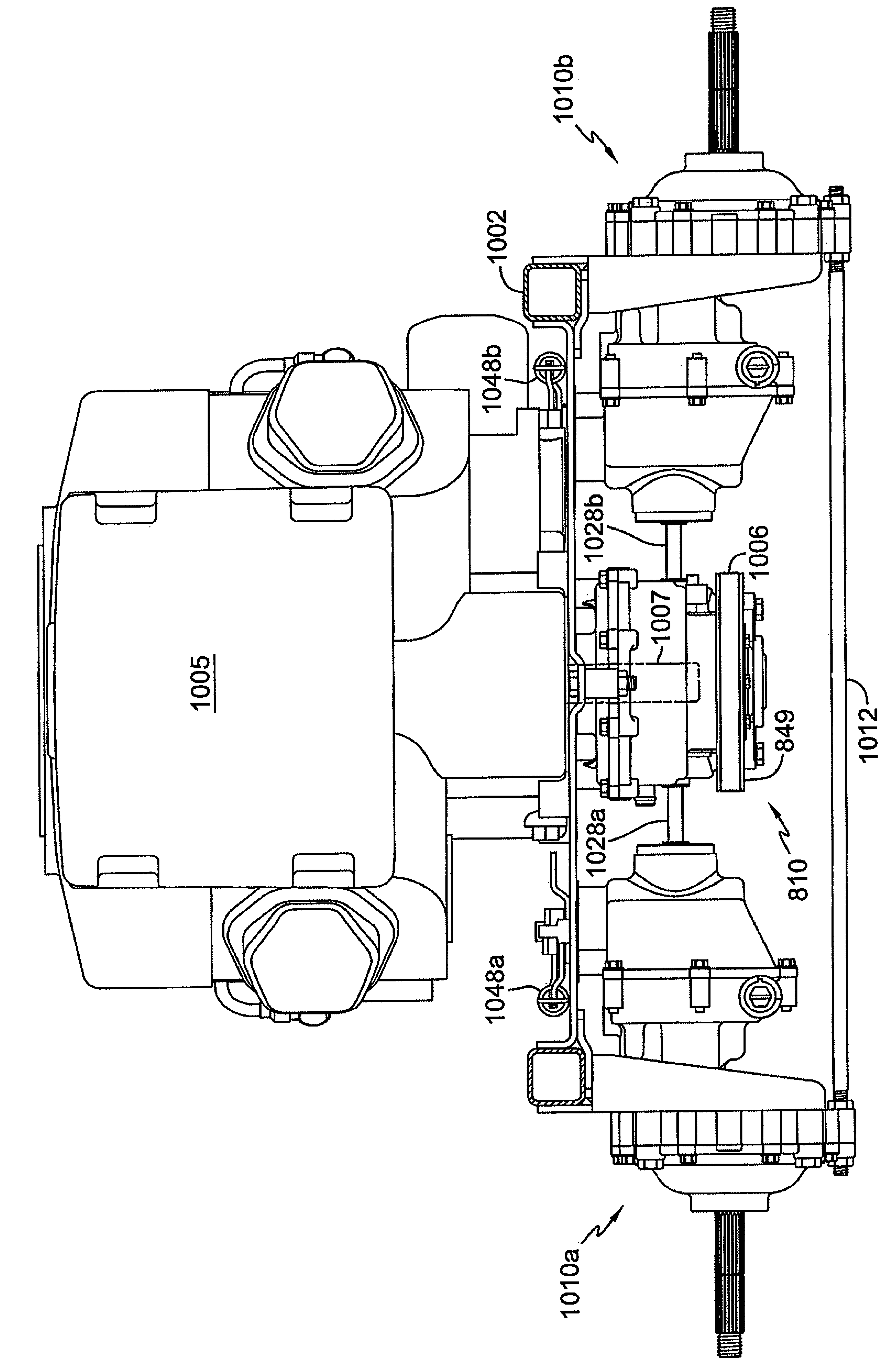
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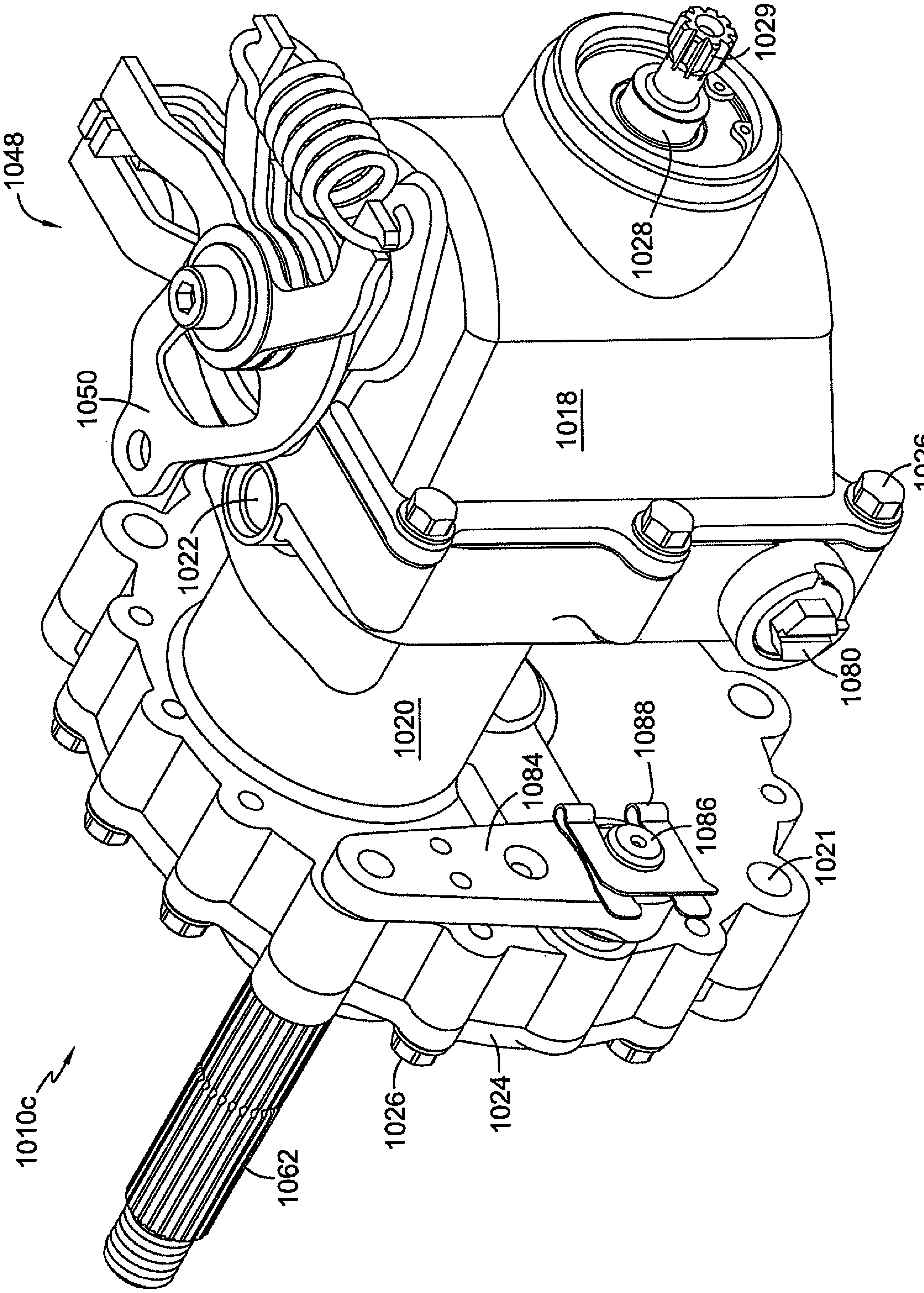
FIG. 29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1021
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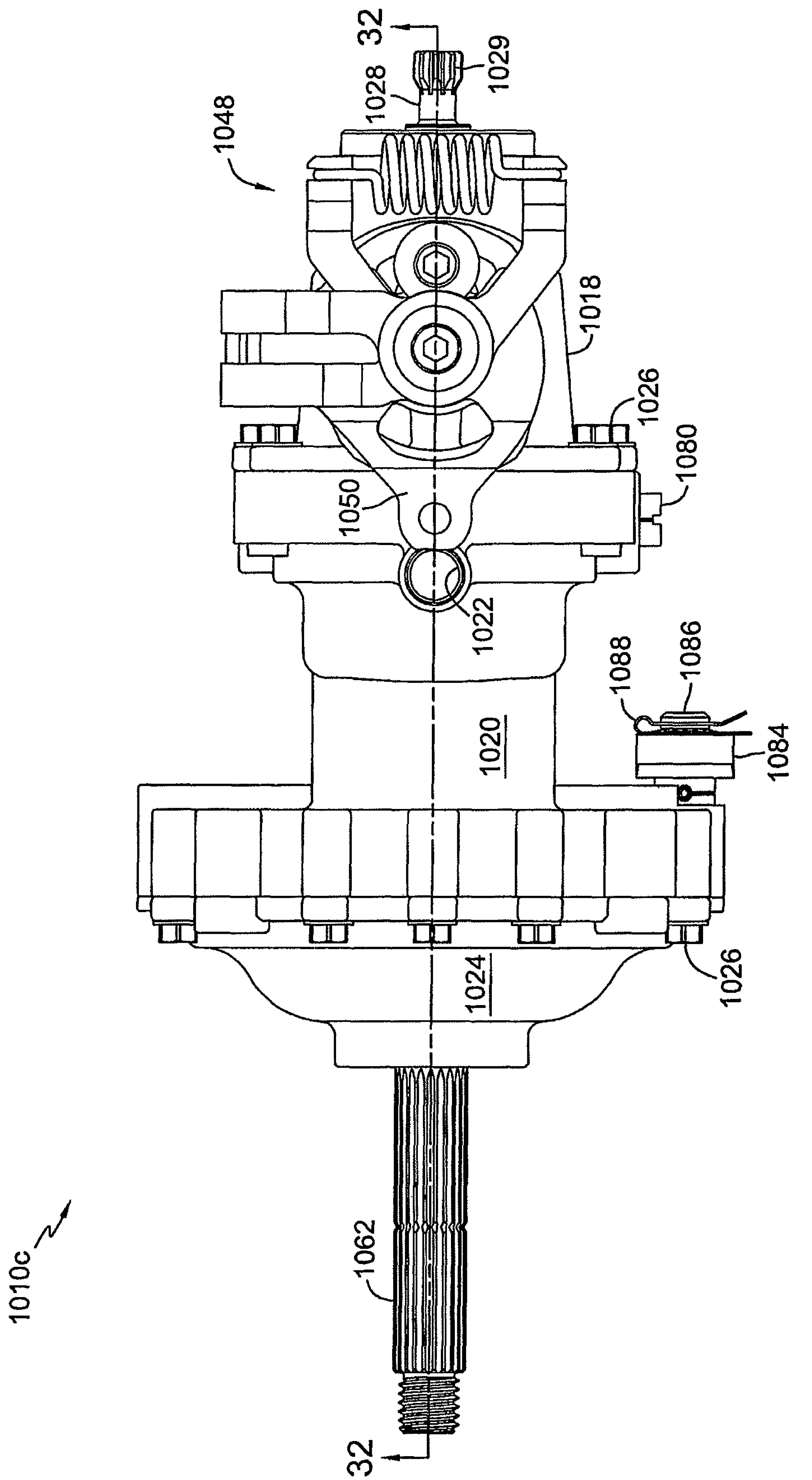
1026
1024

FIG. 30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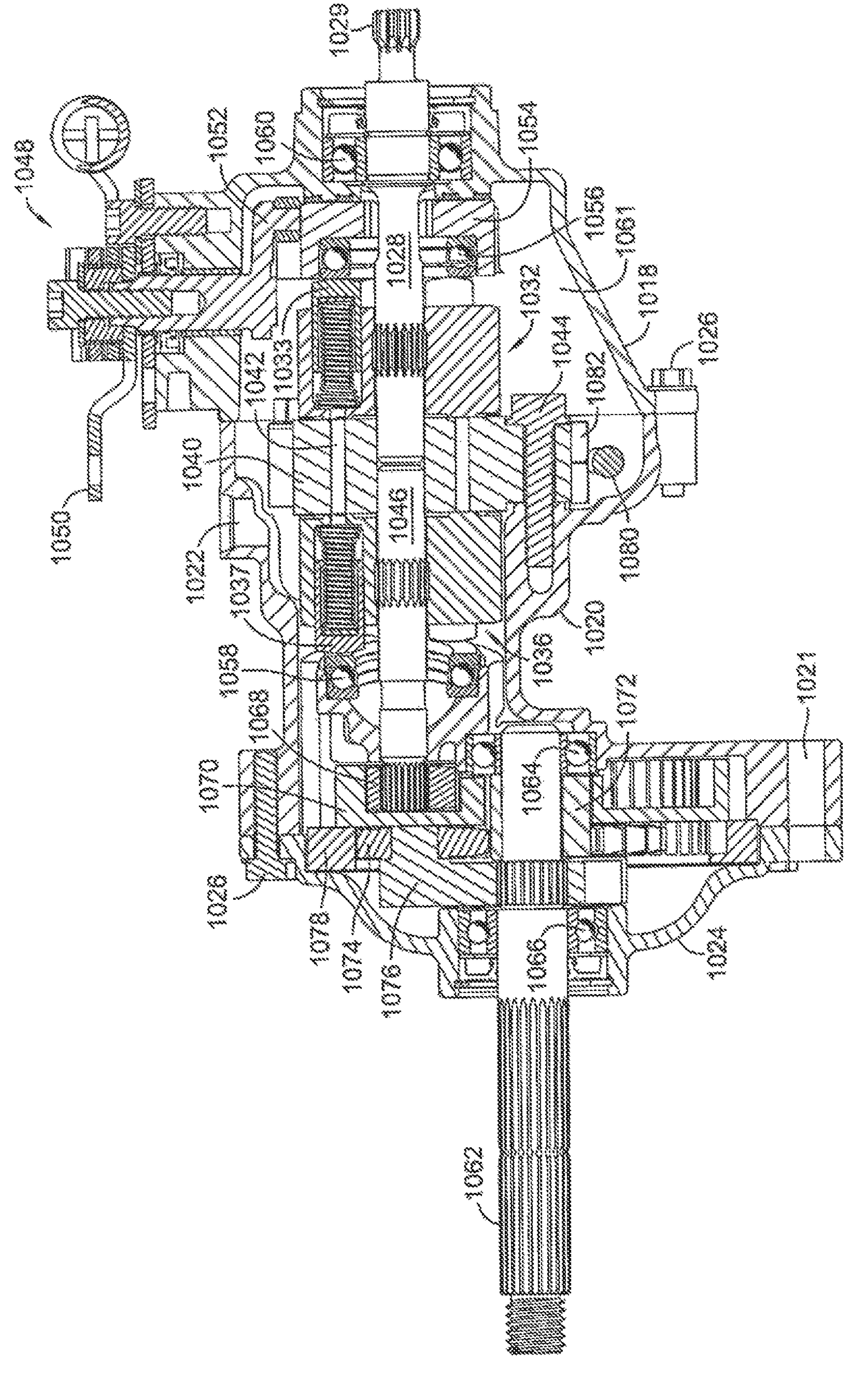
32

FIG. 31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1037
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1024
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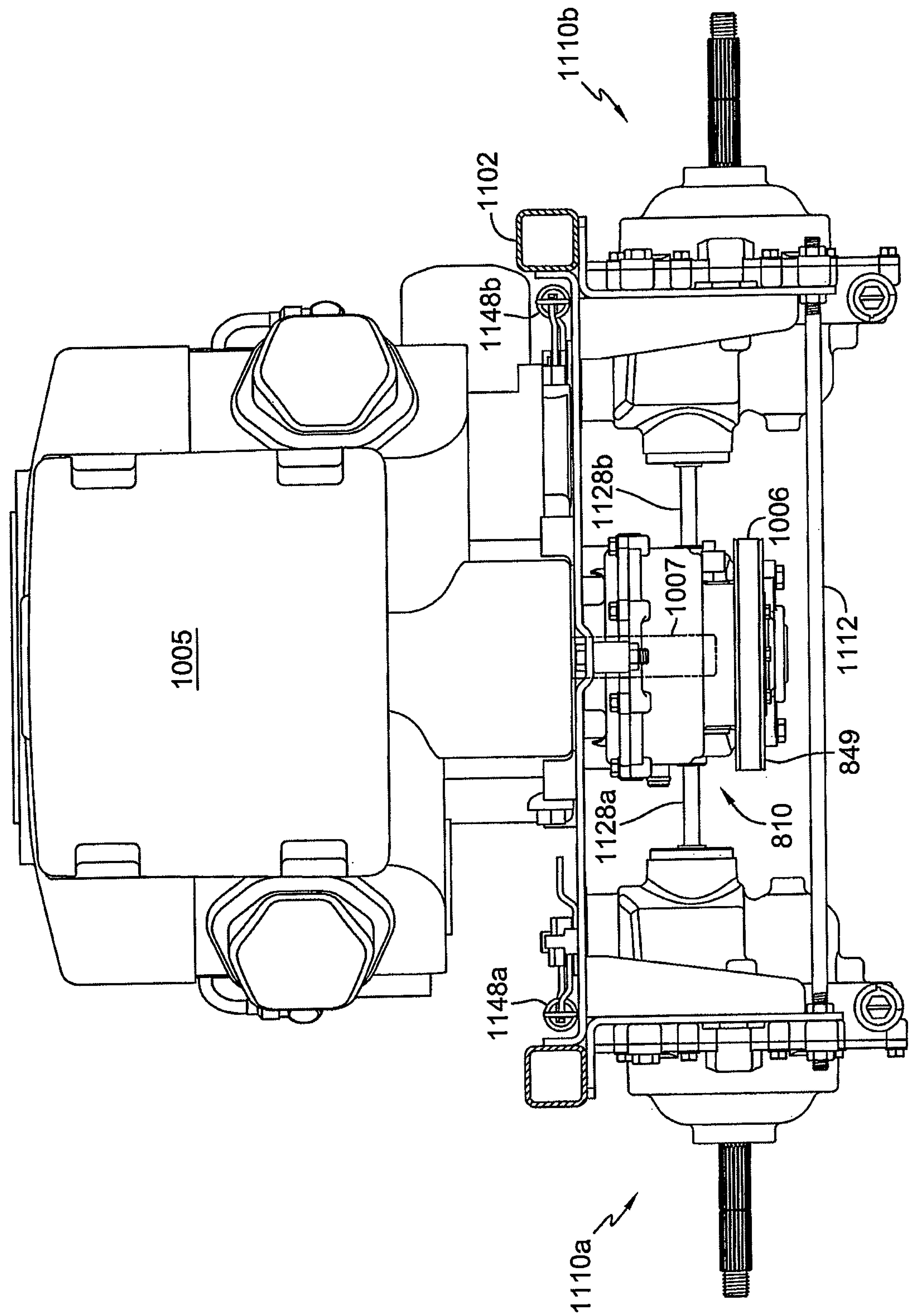
FIG. 32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1148a
1110a

FIG. 33

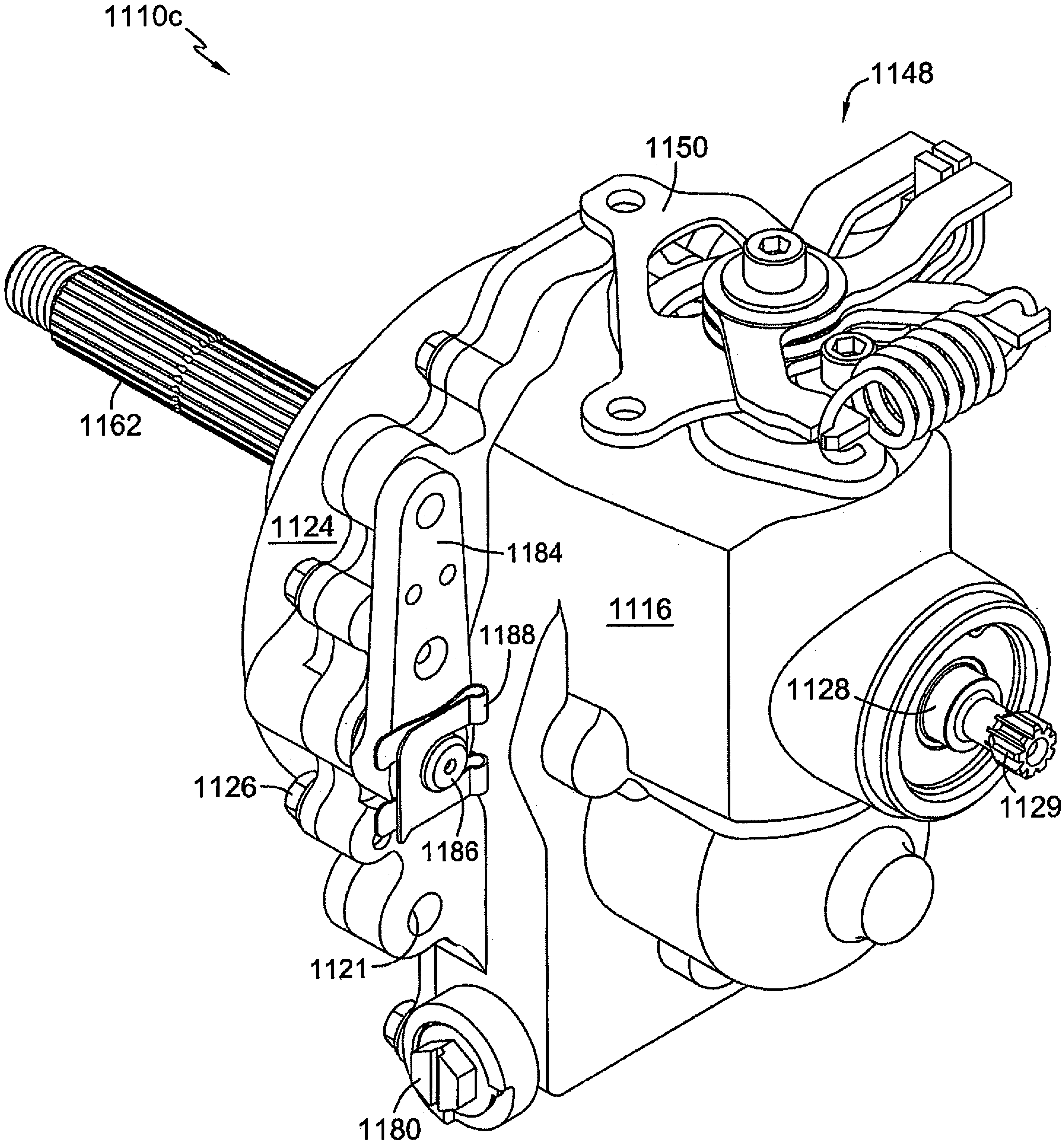
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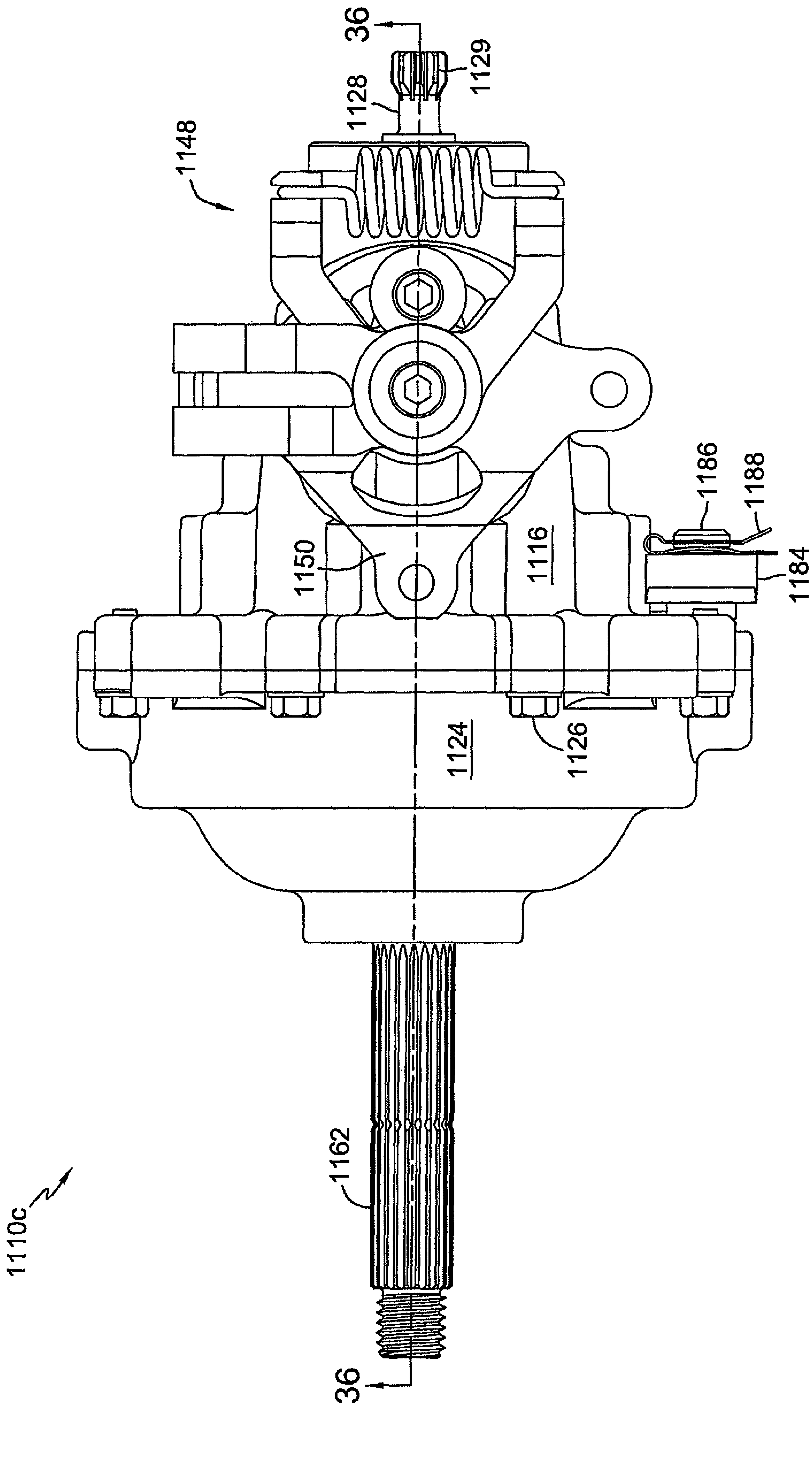
FIG. 34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1126
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1110c
36

FIG. 35

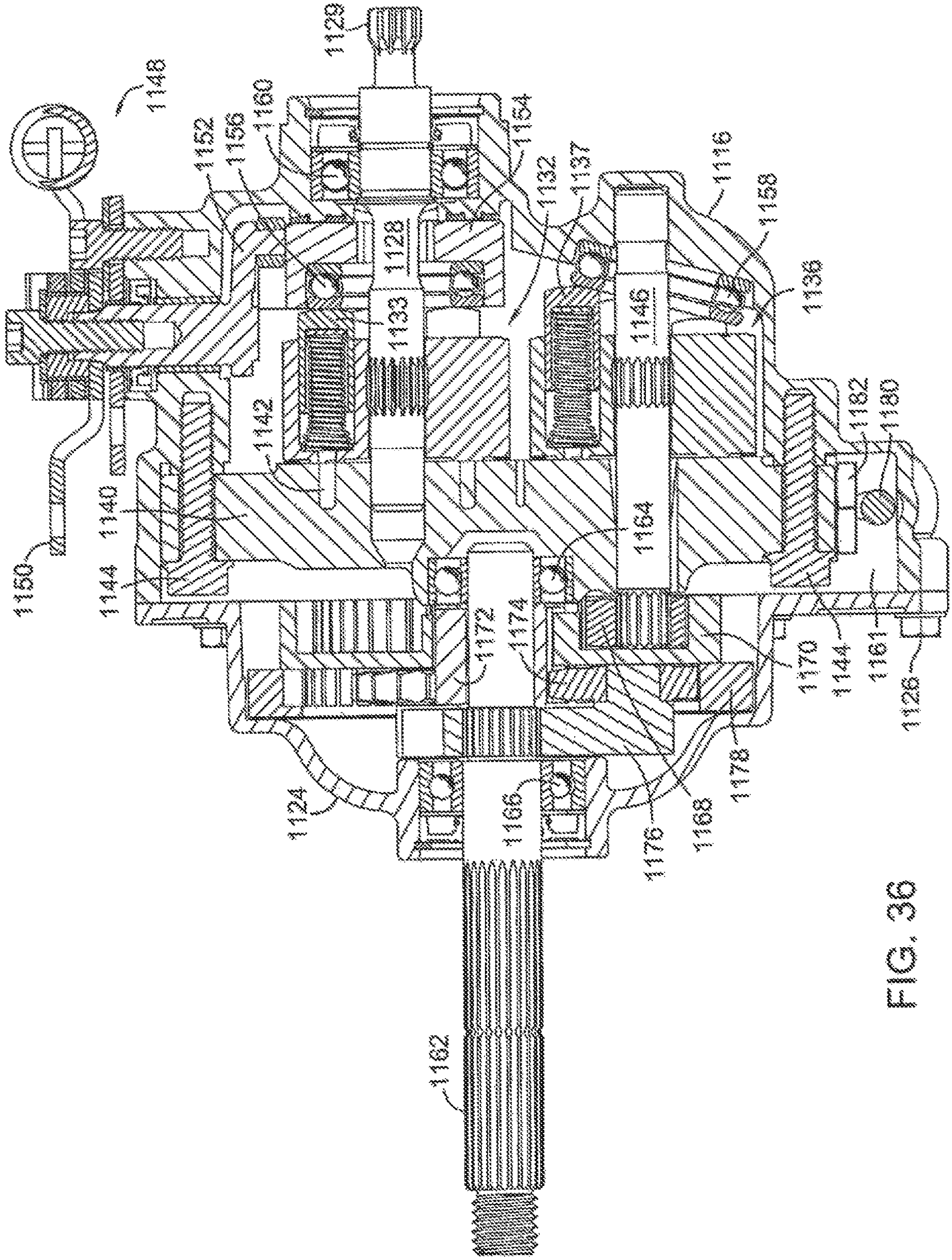
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FIG. 36

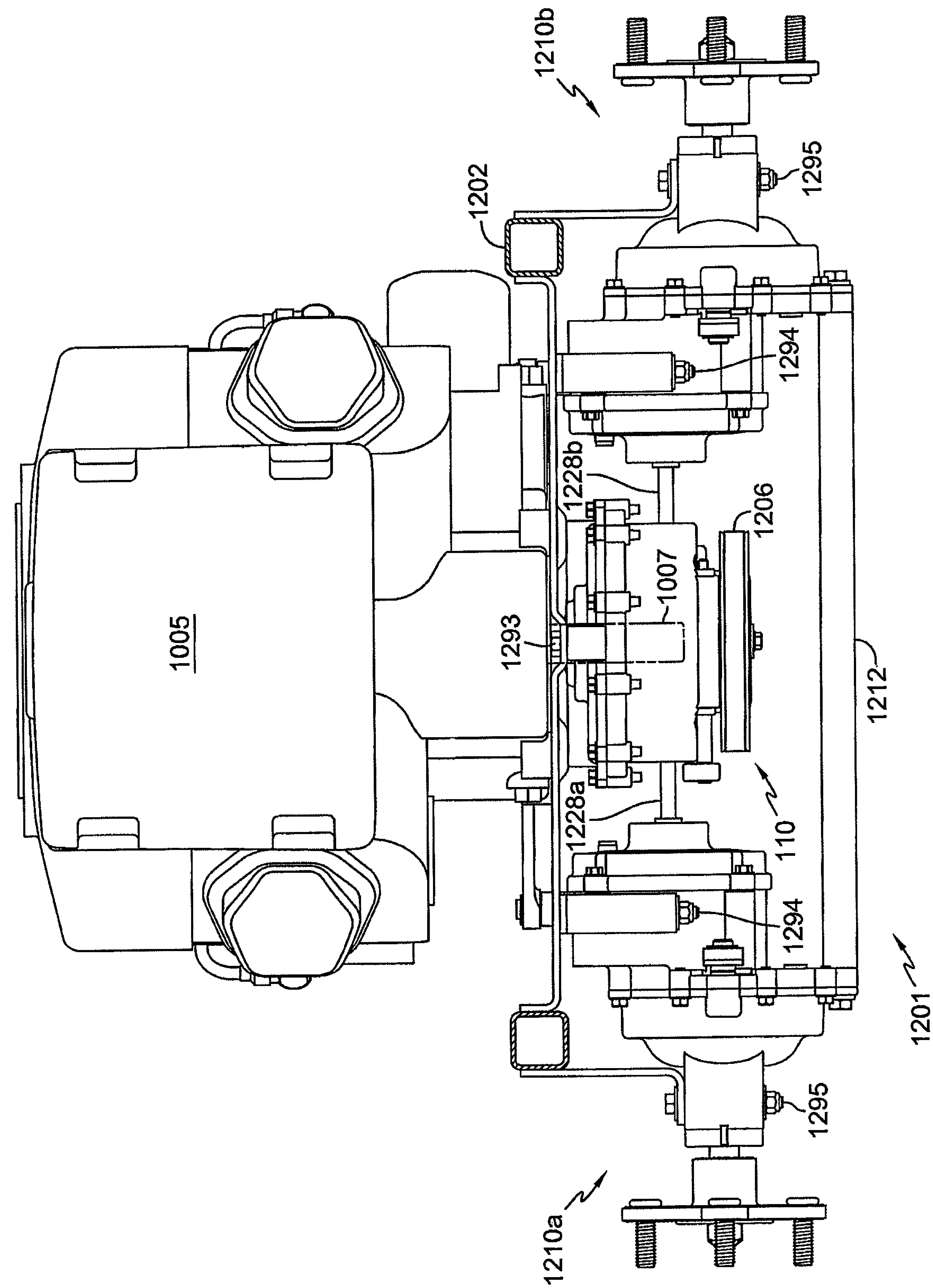
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FIG. 37

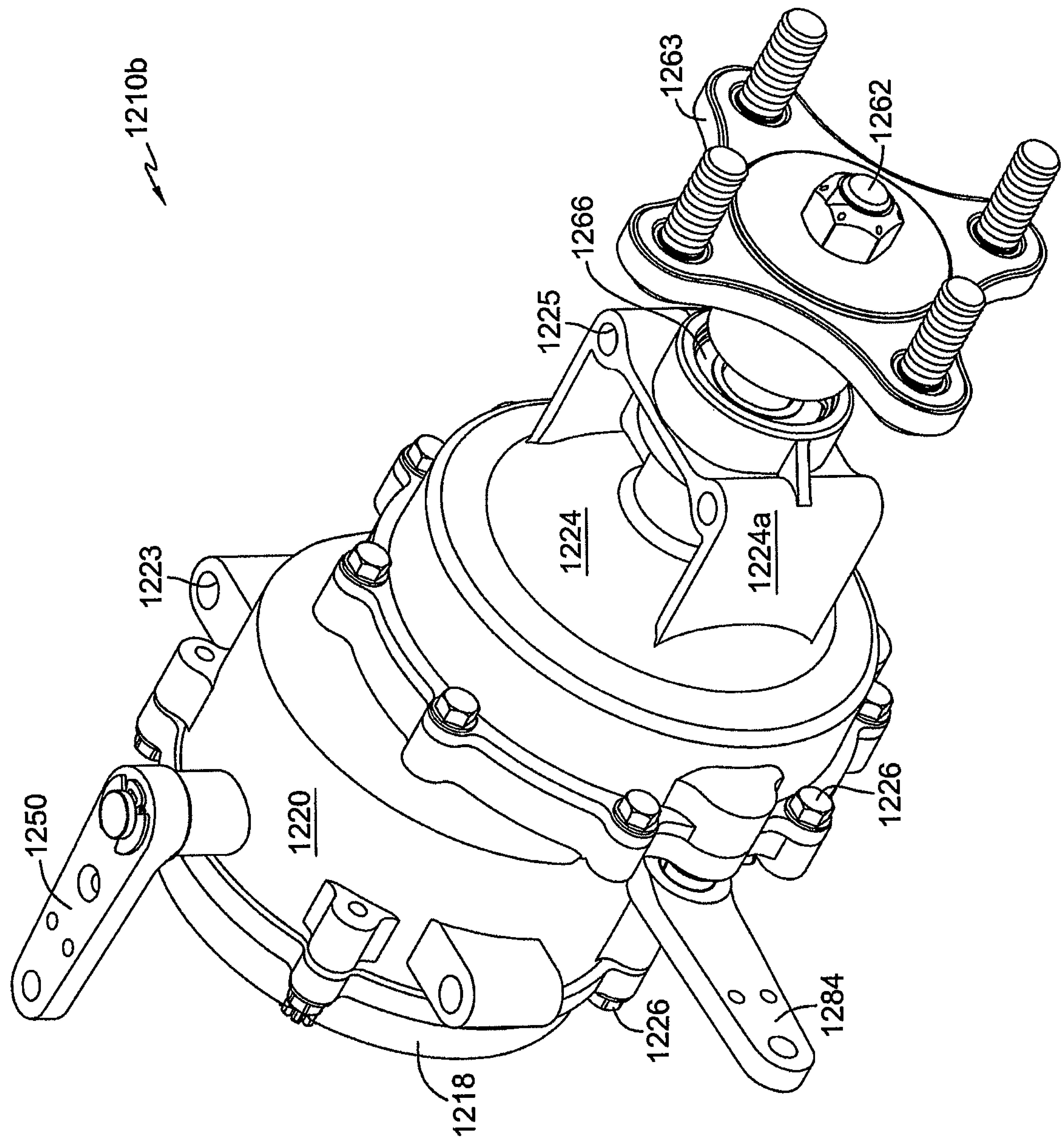
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FIG. 38

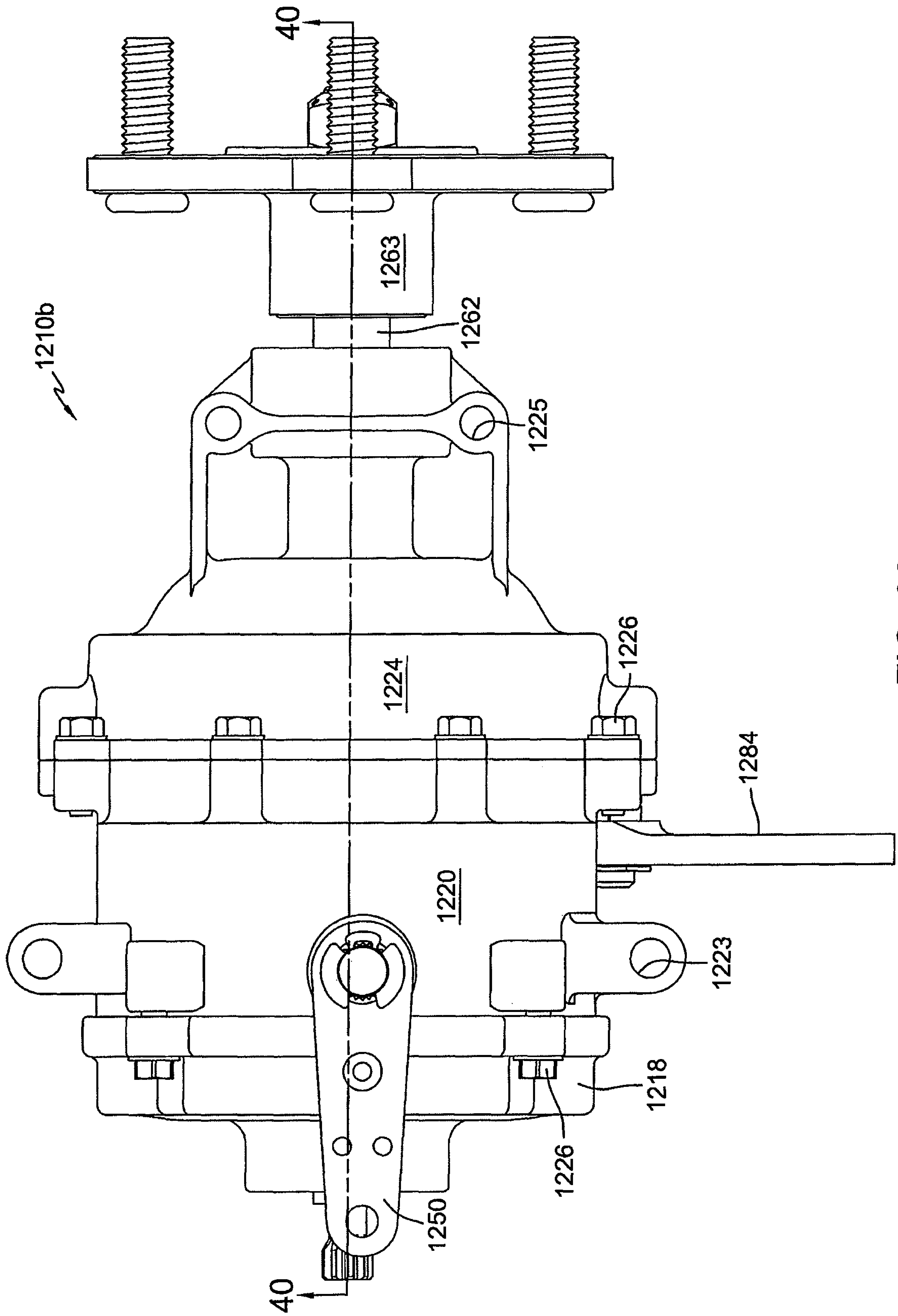
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FIG. 39

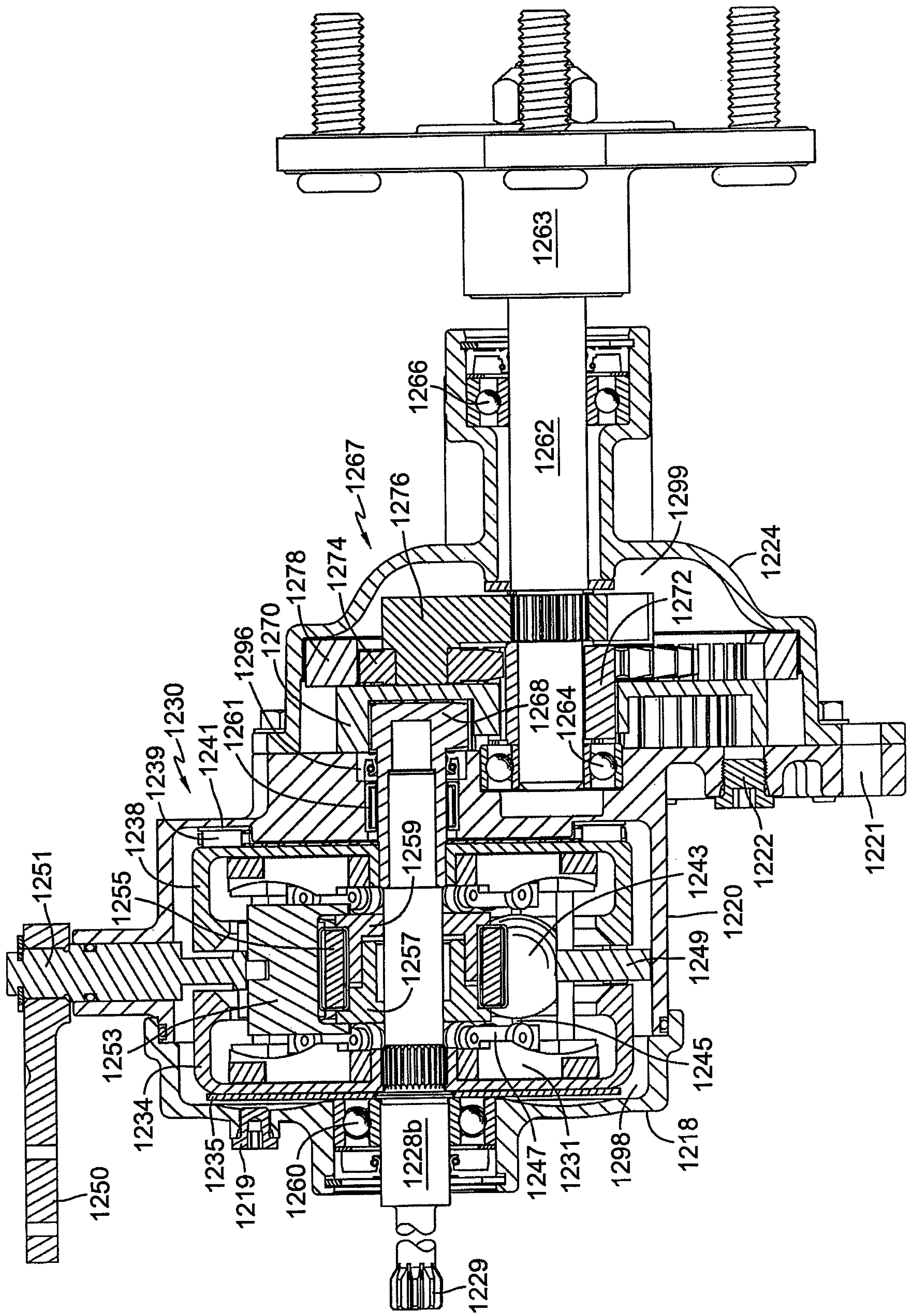
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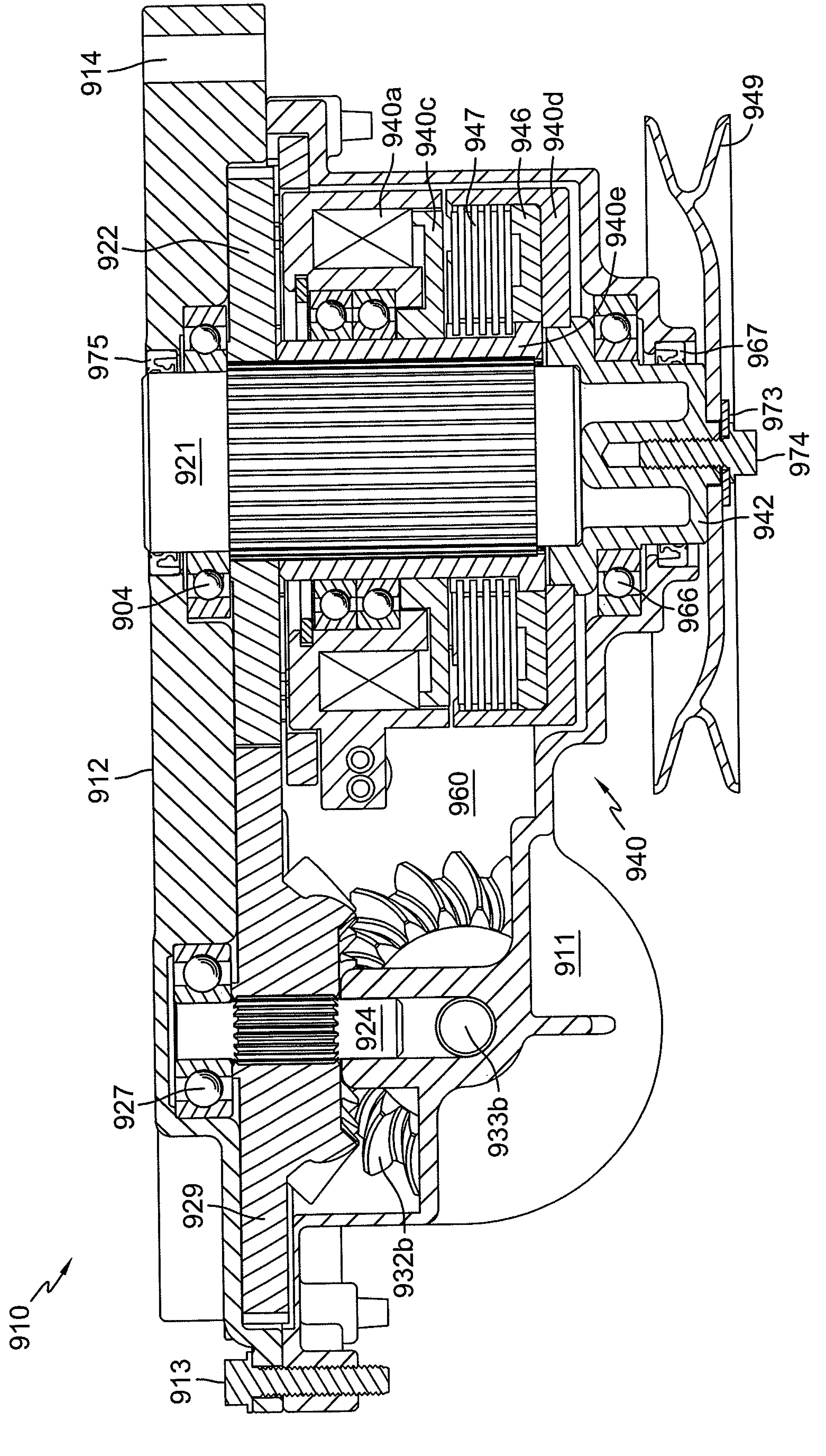
FIG. 40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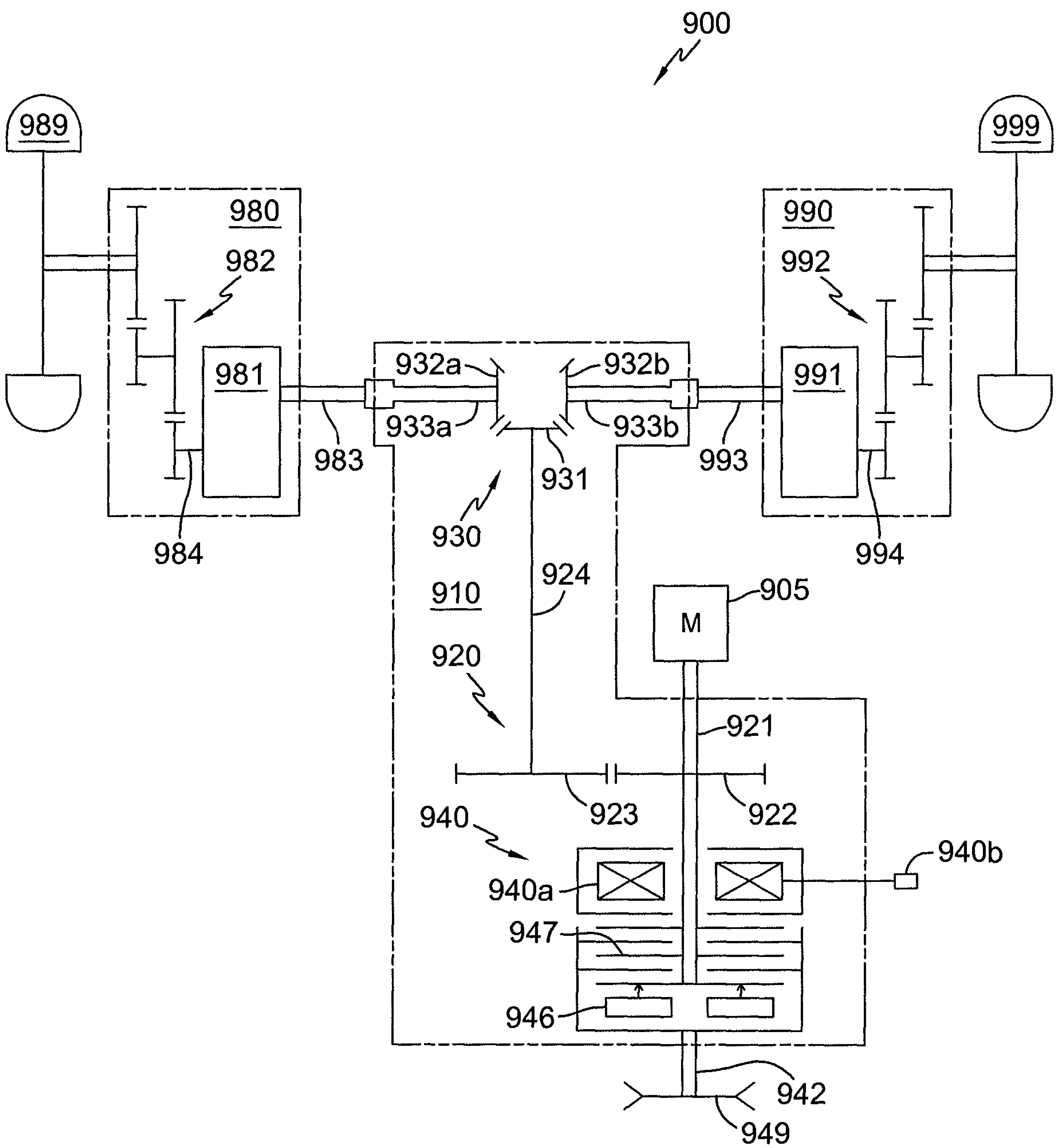
932b
910
913

FIG. 41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930
924
910
905
M
920
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922
940
940b
940a
947
946
942
949

FIG. 42

MODULAR DRIVE SYSTEM

CROSS REFERENCE

This application is a continuation of U.S. patent application Ser. No. 15/985,335, filed on May 21, 2018, which is a divisional of U.S. patent application Ser. No. 14/462,868, filed on Aug. 19, 2014, which is a continuation of U.S. patent application Ser. No. 13/732,928, filed on Jan. 2, 2013, now U.S. Pat. No. 9,194,473, which is a continuation-in-part of U.S. patent application Ser. No. 12/825,038, filed on Jun. 28, 2010, now U.S. Pat. No. 8,393,236, which claims priority from U.S. Provisional Application No. 61/220,795, filed on Jun. 26, 2009; U.S. Provisional Application No. 61/243,097, filed on Sep. 16, 2009; and U.S. Provisional Application No. 61/258,764, filed on Nov. 6, 2009. These prior applications are incorporated herein by reference in their entirety.

BACKGROUND OF THE INVENTION

This application is related to drive systems for zero turn radius (ZT) vehicles. In the past, such drive systems have utilized separate hydraulic pump and wheel motor combinations under individual left and right side control. Such an arrangement permitted the left and right side drive wheels to be rotated in opposite directions to execute zero radius turns about a central point. More recently, integrated transaxles having a hydraulic pump and motor, axle and/or reduction mechanism in a single housing have been used to replace the traditional pump and wheel motor combinations in ZT vehicles. These integrated transaxles have an economy of space and eliminate external closed-loop hydraulic lines. In each of these cases, a prime mover such as an internal combustion engine or electric motor powers the input shafts of the left and right side drive mechanisms through one or more belt and pulley arrangements, leading to some duplication of drive system components.

Most recently, integrated drive mechanisms have been introduced in which the independent left and right side drive mechanisms are combined to form a single drive unit having a central input shaft driven by the prime mover, whether indirectly by belt and pulley arrangement or directly by coupled shaft. Such an arrangement eliminates duplication of the input means between the prime mover and the left and right side drive mechanisms. Typically, the drive mechanisms are hydraulic, but alternative drive mechanisms such as toroidal drives have been used to motivate ZT vehicles. Integrated drive units featuring these various drive mechanisms will have unique design requirements. The rigid structures of such integrated drive units, because they typically span the width of a ZT vehicle's frame, are subject to various loads including torsional loads as the vehicle's frame twists and flexes during severe operation. There exists the need to have a more flexible drive apparatus, one that communicates with the prime mover through a central input shaft, and one that is capable of encompassing a host of drive and output options.

SUMMARY OF THE INVENTION

A modular power distribution and drive system (modular drive system) is provided having a central gear box containing a central drive train and power take off mechanism, wherein various left and right side drive mechanisms may be powered by the central drive train in "plug-and-play" fashion. The left and right side drive mechanisms may be flexibly joined to the central gear box to negate the effect of vehicle frame flexion on the modular drive system. The left and right side drive mechanisms may be of any type requiring input from a prime mover including, but not limited to, hydraulic, toroidal, friction or mechanical drives. And depending upon the configuration of the central drive train, the left and right side drives may be identical or mirror-image units.

The power take off mechanism contains a clutch that may be manual, hydraulic, or electric in design. Where the power take off mechanism contains an electric clutch assembly, the various embodiments of the modular drive system may locate the electric clutch external to the housing of the central gear box, or may locate it internal thereto, e.g. utilizing a wet clutch disposed in the sump of the central gear box.

In some embodiments, the central drive train may also be selectively engaged by a clutch. When the clutch is hydraulic in design, the central gear box may contain a gear pump or charge pump to provide pressurized hydraulic fluid to actuate the clutch. A screw-on filter may be utilized in conjunction with the charge pump to remove debris from the central gear box's hydraulic fluid. A vehicle, such as a riding lawn mower, incorporating such a modular drive system is also provided.

A better understanding of the objects, advantages, features, properties and relationships of the invention will be obtained from the following detailed description and accompanying drawings which set forth illustrative embodiments that are indicative of the various ways in which the principles of the invention may be employed.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic representation of a first embodiment of the modular drive system having a manual clutch and brake.

FIG. **1**A is an isometric view of a representative central gear box in accordance with the first embodiment of the invention.

FIG. **1**B is a top view of the central gear box of FIG. **1**A.

FIG. **1**C is a section view of the central gear box of FIG. **1**B along the line **1**C-**1**C.

FIG. **1**D is a section view of the central gear box of FIG. **1**B along the line **1**D-**1**D.

FIG. **2** is a schematic representation of a second embodiment of the modular drive system having an electric clutch and brake.

FIG. **2**A is an isometric view of a central gear box in accordance with the second embodiment of the invention.

FIG. **2**B is a top view of the central gear box of FIG. **2**A.

FIG. **2**C is a section view of the central gear box of FIG. **2**B along the line **2**C-**2**C with a representative power take off (electric) shown in phantom.

FIG. **3** is a schematic representation of a third embodiment of the modular drive system having a hydraulic clutch and brake.

FIG. **4** is an isometric view of a representative central gear box in accordance with the teachings of FIG. **3**.

FIG. **5** is a top view of the central gear box of FIG. **4** with various elements of the charge gallery and accumulator visible.

FIG. **6** is a section view of the central gear box of FIG. **5** along the line **6**-**6**.

FIG. **7** is a partial section view of the central gear box of FIG. **5** along the line **7**-**7** rotated 90° counterclockwise with some elements shown whole for clarity.

FIG. **8** is a schematic representation of a fourth embodiment of the modular drive system, having a dedicated charge pump to provide pressurized hydraulic fluid to the hydraulic clutch and brake.

FIG. **9** is an isometric view of a representative central gear box in accordance with the teachings of FIG. **8**.

FIG. **10** is a top view of the central gear box of FIG. **9** with various elements of the charge gallery and accumulator visible.

FIG. **11** is a section view of the central gear box of FIG. **10** along the line **11-11**.

FIG. **12** is a schematic representation of a fifth embodiment of the modular drive system, similar to that of FIG. **8**, having an auxiliary pump.

FIG. **13** is a schematic representation of a sixth embodiment of the modular drive system having two hydraulic clutches.

FIG. **14** is an isometric view of a representative central gear box in accordance with the teachings of FIG. **13**.

FIG. **15** is a top view of the central gear box of FIG. **14** with various elements of the charge gallery and accumulator visible.

FIG. **16** is a section view of the central gear box of FIG. **15** along the line **16-16**

FIG. **17** is a partial section view of the central gear box of FIG. **15** along the line **17-17** rotated 90° clockwise with certain elements shown whole for clarity.

FIG. **18** is a schematic representation of a seventh embodiment of the modular drive system having dual hydraulic clutches.

FIG. **19** is an isometric view of a representative central gear box in accordance with the teachings of FIG. **18**.

FIG. **20** is a top view of the central gear box of FIG. **19** with various elements of the charge gallery and accumulator visible.

FIG. **21** is a section view of the central gear box of FIG. **20** along the line **21-21**.

FIG. **22** is an isometric view of another representative central gear box in accordance with the principles of the invention.

FIG. **23** is an isometric view of the central gear box of FIG. **22** rotated one hundred-eighty degrees about the axis of the screw-on filter.

FIG. **24** is a bottom view of the central gear box of FIG. **22**.

FIG. **25** is a partial section view of the central gear box of FIG. **24** along the line **25-25** with some elements shown whole for clarity.

FIG. **26** is a partial section view of the central gear box of FIG. **24** along the line **26-26** rotated 90° counterclockwise with some elements shown whole for clarity.

FIG. **27** is a partial section view of the central gear box of FIG. **24** along the line **27-27** rotated 90° counterclockwise with some elements shown whole for clarity.

FIG. **28** is a bottom view of a vehicle integrating a representative modular drive system, shown here with the central gear box of FIGS. **22-27** and left and right side drive mechanisms similar to the hydrostatic transaxles of FIGS. **30-32**.

FIG. **29** is a rear elevation view of the vehicle of FIG. **28** (with its frame sectioned and elements removed for clarity).

FIG. **30** is an isometric view of a hydrostatic transaxle similar to the right side drive mechanism of the representative modular drive system of FIGS. **28** and **29**.

FIG. **31** is a top view of the hydrostatic transaxle of FIG. **30**.

FIG. **32** is a partial section view of the hydrostatic transaxle of FIG. **31** along the line **32-32** with some elements shown whole for clarity.

FIG. **33** is a rear elevation view of a vehicle integrating a representative modular drive system, shown here with the central gear box of FIGS. **22-27** and left and right side drive mechanisms similar to the hydrostatic transaxles of FIGS. **34-36**.

FIG. **34** is an isometric view of a hydrostatic transaxle similar to the right side drive mechanism of FIG. **33**.

FIG. **35** is a top view of the hydrostatic transaxle of FIG. **34**.

FIG. **36** is a partial section view of the transaxle of FIG. **35** along the line **36-36** with some elements shown whole for clarity.

FIG. **37** is a rear elevation view of a vehicle (with its frame sectioned and elements removed for clarity) integrating a representative modular drive system, shown here with the central gear box of FIGS. **1A-1D** and a pair of toroidal left and right side drive mechanisms.

FIG. **38** is an isometric view of the right side drive mechanism of FIG. **37**.

FIG. **39** is a top view of the right side drive mechanism of FIG. **38**.

FIG. **40** is a section view of the right side drive mechanism of FIG. **39** along the line **40-40**.

FIG. **41** is a section view of another central gear box in accordance with the principles of the invention, the section being similar to that depicted in FIG. **2C**, but having an electric clutch disposed in the sump of the central gear box.

FIG. **42** is a schematic representation of an alternative embodiment of the modular drive system incorporating the central gear box shown in FIG. **41**, absent the compact variation to its gear train, i.e. the combination gear.

DETAILED DESCRIPTION OF THE DRAWINGS

The description that follows describes, illustrates and exemplifies one or more embodiments of the present invention in accordance with its principles. This description is not provided to limit the invention to the embodiments described herein, but rather to explain and teach the principles of the invention in order to enable one of ordinary skill in the art to understand these principles and, with that understanding, be able to apply them to practice not only the embodiments described herein, but also other embodiments that may come to mind in accordance with these principles. The scope of the present invention is intended to cover all such embodiments that may fall within the scope of the appended claims, either literally or under the doctrine of equivalents.

It should be noted that in the description and drawings, like or substantially similar elements may be labeled with the same reference numerals. However, sometimes these elements may be labeled with differing numbers, such as, for example, in cases where such labeling facilitates a more clear description. Additionally, the drawings set forth herein are not necessarily drawn to scale, and in some instances proportions may have been exaggerated to more clearly depict certain features. Such labeling and drawing practices do not necessarily implicate an underlying substantive purpose. As stated above, the present specification is intended to be taken as a whole and interpreted in accordance with the principles of the present invention as taught herein and understood by one of ordinary skill in the art.

FIG. **1** illustrates a first embodiment of a modular drive system **100** in accordance with the principles of the invention. In general, modular drive system **100** comprises a central gear box **110** configured to be driven by a prime mover **105**, which distributes the motive force received therefrom to a pair of independent drive mechanisms, left side drive mechanism **180** and right side drive mechanism **190** respectively. Modular drive system **100** may be powered by a prime mover **105**, such as the depicted internal combustion engine M or an electric motor, through input shaft **121**. Though illustrated as a direct drive configuration, it should be understood that prime mover **105** may also transfer power to input shaft **121** indirectly by a belt and pulley arrangement.

The central gear box **110** comprises a central drive train having an input shaft **121**, a transmission gear set **120**, an intermediate or jack shaft **124**, a distribution gear set **130**, and a central output shaft **133**. Input shaft **121** drives transmission gear **122**, which in turn drives corresponding transmission gear **123** to power jack shaft **124**. Transmission gears **122** and **123** may be spur gears or helical gears depending on the relative need of the modular drive system **100** for efficiency, revolutions per minute (rpm) and noise reduction. Jack shaft **124** provides motive force to distribution gear set **130**, represented here as bevel gears **131** and **132**, and ultimately to central output shaft **133**. It should be understood that bevel gears **131** and **132** may similarly be replaced by spiral bevel gears where higher rpm and/or greater noise reduction is needed. Central output shaft **133** is illustrated with splined female ends **134** and **135** to accommodate input shafts **183** and **193** from the left and right side drive mechanisms **180** and **190** respectively. The inverse configuration (not shown) could also be utilized, wherein input shafts **183** and **193** exhibit the female splined connection. The drive mechanisms **180** and **190** may be flexibly joined to central gear box **110** through the mating of coarse toothed shaft ends, male and female, wherein the male shaft end has a rounded or tapered profile. Such a flexible joint negates the effect of vehicle frame flexion on the modular drive system. It should be understood that other flexible joints known in the art, such as universal or Cardan joints, are contemplated within the scope of the invention. Regardless of joint configuration, central gear box **110** can accommodate a variety of drive mechanisms in "plug-and-play" fashion provided the appropriate input shaft is selected. Additionally, the length of input shafts **183** and **193** can be varied to accommodate differences in vehicle frame widths.

Left and right side drive mechanisms **180** and **190** consist of variable speed drives **181** and **191**, respectively, which are independently controlled by a vehicle operator through external linkages (not shown). As previously indicated, drives **181** and **191** may be of any type requiring input from a prime mover including, but not limited to, hydraulic, toroidal, friction or mechanical drives. Because the rotation of input shafts **183** and **193** are identical, variable speed drives **181** and **191** are preferably mirror-image units to translate the same rotation to output shafts **184** and **194** respectively, if the external left and right side linkages (not shown) are to move the respective actuators (not shown) of the variable speed drives **181** and **191** in the same rotational direction (e.g. toward the forward end of a vehicle to move forward). While such symmetric control motions are desirable, it is to be understood that identical variable speed drives may be utilized on the left and right sides to produce the same rotation at output shafts **184** and **194**, with actuators being moved in opposite rotational directions, if the variable speed drives are capable of both forward and reverse stroking, such as with hydraulic drives for example.

Reduction gear sets **182** and **192** multiply the torque and reduce the speed of the output to drive axles **185** and **195**, providing power to drive wheels **189** and **199** respectively. Though illustrated as two-stage, spur gear reductions, it should be understood that reduction gear sets **182** and **192** may include any reduction mechanisms known in the art, including but not limited to single or double stage planetary reductions.

Central gear box **110** further comprises a power take off mechanism having a manual clutch and brake assembly **140** which is also powered by input shaft **121**. Spring **143** normally biases the brake to the engaged position, wherein brake plates **164** are brought into frictional engagement by action of a cam plate **146** or pins (not shown). An operator engages the clutch through actuation of lever **148** which disengages the brake and brings the clutch plates **147** of cage **141**, which rotate in unison with input shaft **121**, and power take off output shaft **142** into frictional engagement. As a result, input shaft **121** and power take off output shaft **142** rotate as a unit to drive an output device such as pulley **149**. On a ZT vehicle, pulley **149** may be used to drive auxiliary devices such as a mowing deck or snow thrower. While the inclusion of a brake on the power take mechanism has safety advantages for the vehicle operator and will be discussed with respect to the embodiments herein, it is to be understood that the invention contemplates a power take off mechanism having a clutch alone.

FIGS. **1A-1D** depict a representative central gear box **110**. Gear box housing **111** contains the central drive train and power take off mechanism. Gear box housing cover **112** is sealingly engaged to housing **111** by a plurality of fasteners **113** to form sump **160**, which is filled with oil via a port (not shown) on the side of housing **111**. Housing cover **112** is formed with a plurality of bosses **114**, by which central gear box **110** may be fastened to a vehicle frame. Input shaft **121** serves as both the input shaft for the central drive train and the power take off mechanism. In FIGS. **1A** and **1B**, input shaft **121** is depicted as a hollow shaft with multiple slots or keyways to receive the output shaft of a prime mover, such as output shaft **1007** of internal combustion engine **1005** in FIG. **37**. The output of the central drive train comprises a pair of central output shafts **133*a*** and **133*b***, which rotate in opposite senses. These outputs power the left and right side drive mechanisms, respectively, of a modular drive system in accordance with the principles of the invention. It will be understood, however, that a single, central output shaft may be used to drive both the left and right side drive mechanisms in the same rotational sense. As depicted in FIG. **1A**, central output shaft **133*a*** comprises a coarse tooth, female splined end, which permits a modicum of flexibility in the driveline of the modular drive system when mated to the input shaft of an outboard drive mechanism having a corresponding structure bearing a somewhat curved or tapered profile. Such an input shaft end **1229** is best illustrated in FIG. **40**. These characteristics are desirable in a utility vehicle such as a ZT mower wherein the vehicle frame is subject to flexion.

External to gear box housing **111** are portions of power take off actuator **148**, comprising actuator arm **148*a*** and actuator shaft **148*b***. When engaged, power take off mechanism **140** provides motive force to pulley **149**, which in turn may be used to indirectly drive various mechanical implements such as a mowing deck through use of a v-belt. It be noted that the narrow width of gear box housing **111** proximate to central output shafts **133*a*** and **133*b*** permits routing of the v-belt adjacent thereto.

FIGS. 1C and 1D detail the inner workings of central gear box **110**. Input shaft **121** is rotationally supported in central gear box **110** by bearing **104**, located in gear box housing cover **112**, and bushing **107** located interior to power take off output shaft **142**, which in turn is supported by bearing **166** in gear box housing **111**. Input shaft **121** has an external gear form or spline **121***a* that engages and drives both transmission gear **122** of the central drive train and the outer clutch plate carrier **141***d* of power take off mechanism **140**.

Transmission gear **122** engages and drives transmission gear **129**, which rotates upon an intermediate or jack shaft **124**, which is supported at a first end by a bearing **127** located in gear box housing cover **112**, and journaled at a second end by gear box housing **111**. Transmission gear **129** comprises a second gear form **129***a*, in this instance a bevel gear form that engages a complementary bevel gear **132***b* to drive central output shaft **133***b*. This is a compact variation on the gear train depicted in FIG. **1**, wherein gear form **129***a* replaces bevel gear **131**, a second, individual gear splined to jack shaft **124**. Central output shaft **133***b* may be journaled at a first, inboard end by gear box housing **111** and supported by an outboard bearing or the like (not shown) as known in the art. As discussed earlier, a parallel structure comprising a second bevel gear (not shown) driven by second gear form **129***a* is splined to central output shaft **133***a*, providing a second rotational output for the central drive train. As illustrated in FIG. **1**C, second gear form **129***a* and bevel gear **132***b* comprise spiral bevel gear forms, which are suitable for higher revolutions per minute (rpm) and/or greater noise reduction applications as compared to standard bevel gears. It will be understood, however, that both standard and spiral bevel gears are contemplated within the scope of the invention. Similarly, transmission gears **122** and **129** may be spur gears (as illustrated) or helical gears depending on the relative need of the modular drive system for efficiency, rpm, and/or greater noise reduction.

Power take off mechanism **140** comprises both a clutch assembly and a brake assembly. Power take off actuator **148** controls both of these assemblies. When actuator arm **148***a* is not rotated into engagement, a spring assembly **143** biases the power take off output shaft **142** to a braked condition and prevents engagement of the clutch assembly. When actuator arm **148***a* is rotated into engagement against the force of spring assembly **143**, the brake assembly is first released followed by engagement of the clutch assembly, whereby the rotation of input shaft **121** is imparted to power take off output shaft **142**.

In FIGS. **1**C and **1**D, the clutch assembly is illustrated as engaged and the brake assembly is shown as disengaged. More specifically, actuator arm **148***a* has been rotated clockwise about the axis of actuator shaft **148***b* to which it is fixed. Actuator shaft **148***b* correspondingly rotates actuator fork **148***c* upward against washer **148***d*, which lies concentric about power take off output shaft **142**. Washer **148***d* pushes upward against thrust bearing **187***b* which also lies concentric about power take off output shaft **142**. Inner clutch plate carrier **141***e* is situated upon thrust bearing **187***b* and is slidably engaged to an outer gear form on power take off output shaft **142** via a corresponding gear form on the inner clutch plate carrier's inner diameter. The upward movement of thrust bearing **187***b* acts to push inner clutch plate carrier **141***e* upward against lower spring assembly washer **143***c*, compressing spring **143***a* against upper spring assembly washer **143***b* and thrust bearing **187***a*. The components of spring assembly **143** and thrust bearing **187***a* lie concentric to input shaft **121** and are trapped between outer clutch plate carrier **141***d* and power take off output shaft **142**.

Engaged to a slot on the outer surface of inner clutch plate carrier **141***e* is a retaining ring **144** that serves to act against either thrust washer **147***a* to engage the clutch assembly or thrust washer **164***a* to engage the brake assembly as the inner clutch plate carrier **141***e* moves vertically up and down. A stack-up of clutch plates **147** is alternatively retained by a plurality of fingers **141***c* on outer clutch plate carrier **141***d* to rotate therewith and a gear form on the outer surface of inner clutch plate carrier **141***e*. An upward movement of inner clutch plate carrier **141***e* forces retaining ring **144** against thrust washer **147***a* to place clutch plates **147** into frictional engagement. Power take off output shaft **142** is thereby rotated synchronously with input shaft **121**. A stack-up of brake plates **164** is alternatively retained by a plurality of fingers **111***a* formed in gear box housing **111** to remain stationary therewith and a gear form on the outer surface of inner clutch plate carrier **141***e*. As actuator arm **148***a*, and correspondingly actuator shaft **148***b*, is rotated counterclockwise, actuator fork **148***c* no longer acts vertically against washer **148***d*. Consequently, spring assembly **143** acts to move inner clutch plate carrier **141***e* downward bringing retaining ring **144** into engagement with thrust washer **164***a*, thereby placing brake plates **164** into frictional engagement. As a result, inner clutch plate carrier **141***e* and power take off output shaft **142** are held in a braking condition, preventing the rotation of pulley **149** which is fixedly engaged to power take off output shaft **142** via fastener **174** and washer **173**.

It should be noted that shaft seals **175** and **167** are incorporated to prevent leakage of oil from sump **160** at the exterior ends of input shaft **121** and power take off output shaft **142** respectively. Similarly, shaft seals (not shown) are incorporated to prevent leakage of oil at the openings formed in gear box housing **111** for central output shafts **133***a* and **133***b* to permit external routing of actuator shaft **148***b*.

Successive embodiments of modular drive systems (**200**, **300**, **400**, **500**, **600** and **700**) in accordance with the principles of the invention are schematically depicted in FIGS. **2**, **3**, **8**, **12**, **13** and **18**. In general, these embodiments differ in the type of clutch mechanism incorporated, the configuration of the distribution gear set, the incorporation of a charge pump **459** and **659**, and the presence of an auxiliary pump **558**. An additional embodiment of a central gear box **810** incorporating a bolt-on charge pump **859**, an input shaft **821** that can receive the output shaft of a prime mover at either end, and a screw-on filter **853** is depicted in FIGS. **22** through **27**. The differences among all these embodiments will be explained in detail below. It should be understood that the variation depicted among common elements may be interchangeably combined to create further embodiments in accordance with the principles of the invention. Throughout this specification, common elements among the various embodiments are numerically labeled in sequence, where possible, for easier identification and understanding (e.g. **121**, **221**, **321**, **421**, **521**, **621**, **721**, and **821** all pertain to input shafts). Such common elements will not always be discussed for each embodiment, but will be understood to operate as described in a previous embodiment.

FIG. **2** illustrates a second embodiment of a modular drive system **200**. Here, an electric clutch and brake assembly **240** alternately prevents rotation of power take off output shaft **242** or couples the rotation of input shaft **221** to output shaft **242**. As with assembly **140**, the bias of the brake in assembly **240** is toward engagement, thereby braking the rotation of output shaft **242** and pulley **249**. This is a safety mechanism for use in applications where assembly **240** provides motive force to an auxiliary device such as a mower deck or snow thrower. An operator disengages the brake and engages the clutch of assembly **240** by closing a switch (not shown) conveniently mounted on the ZT vehicle. At this point, electricity from a switchable source of direct current/voltage (not shown) energizes the at least one coil of assembly **240**, thereby attracting the respective armatures to the at least one coil to disengage the brake and synchronize the rotations of input shaft **221** and output shaft **242**. The reverse process occurs when direct current/voltage is removed by opening the switch. This clutch and brake assembly **240**, unlike those of the other embodiments described herein, is external to the oil containing central gear box **210**.

Central gear box **210** contains the same central drive train configuration as that of central gear box **110** and will not be further described herein. The connections between central gear box **210** and the left and right side drive mechanisms **280** and **290** illustrate another means of attachment, wherein a coupling **238** and an intermediate shaft **236** are interposed between the input shaft **283** of left side drive mechanism **280** and the left end **234** of central output shaft **233**. Similarly, coupling **239** and intermediate shaft **237** are interposed between the input shaft **293** of right side drive mechanism **290** and the right end **235** of central output shaft **233**. This configuration allows more flexibility during the assembly process, permitting assemblers to mount the respective drive mechanisms to a vehicle's frame without having to form the couplings simultaneously. And similar to modular drive system **100**, modular drive system **200** can be adjusted to fit various vehicle frame widths, most simply by varying the length of intermediate shafts **236** and **237**.

FIG. **2** also illustrates that modular drive systems in accordance with the principles of the invention may accommodate variable speed drives having an offset configuration. The input shaft **283** and output shaft **284** of variable speed drive **281** are not coaxial as previously seen on variable speed drives **181** and **191**. Similarly, input shaft **293** and output shaft **294** are not coaxial. This offset allows for the production of a more compact vehicle, reducing the overall length of the vehicle. Regardless of offset, central gear box **210** can power any drive mechanism having the appropriate input shaft configuration.

FIGS. **2A-2C** depicts central gear box **210** configured to allow a prime mover's output shaft (not shown) to pass directly through a hollow input shaft **221**, permitting a variety of electric clutch and brake assemblies, generically depicted herein as electric clutch and brake assembly **240**, to be mated to the prime mover's output shaft. It should also be noted that central gear box **210** can be utilized without the electric clutch and brake assembly **240**, which then permits central gear box **210** to be optionally mounted in an inverted position. This orientation would allow the prime mover to be mounted lower in a vehicle, improving the vehicle's center of gravity. As illustrated, input shaft **221** is rotationally supported by a pair of bearings **204** located in gear box housing **211** and gear box housing cover **212** respectively. Similarly, input shaft **221** is sealed at both ends by a pair of shaft seals **275** that prevent leakage of oil from the sump **260** formed by the combination of gear box housing **211** and gear box housing cover **212**. The compact length of input shaft **221** creates a narrow profile to central gear box **210** that accommodates placement of electric clutch and brake assembly **240**. While the internal configuration of input shaft **221** varies from that depicted for input shaft **121** of central gear box **110**, it will be understood that any combination of slots, keys and/or keyways known in the art is contemplated within the principles of the invention.

Rotation of the field back plate of electric clutch and brake assembly **240** can be restrained by any method commonly known in the art such as fixing the plate to a vehicle's frame with an additional flat piece of metal (not shown). Electric clutch and brake assembly **240** acts to engage or arrest the rotation of pulley **249** when an electric current is applied to or removed from the field assembly (not shown). The function of such electric clutch and brake assemblies is well known in the art and will not be further described herein.

The structure and function of the central drive train of central gear box **210** is identical to that previously described for central gear box **110**, including the compact variation created by use of combination gear **229**, and will not be further described herein.

FIGS. **3**, **4**, **5**, **6** and **7** illustrate a third embodiment of a modular drive system **300** wherein transmission gears **322** and **323** form a gear pump **350**. Such an arrangement requires finer machining tolerances for the gears than in previously described embodiments, but provides the benefit of supplying a source of pressurized hydraulic fluid sufficient to actuate a hydraulic clutch and brake assembly **340** in the power take off mechanism. Prime mover **305**, through input shaft **321**, drives gear pump **350**. An o-ring **375** seals the opening for input shaft **321** in gear box housing cover **312**. The intermeshing of transmission gears **322** and **323** in an appropriately configured volume separated from fluid sump **360** by a charge plate **328** (as shown in FIG. **6**) provides pressurized hydraulic fluid to a charge gallery **354** (as best shown in FIG. **5**) in fluid communication with an accumulator **351**.

FIG. **6** further illustrates a compact variation on the central drive train illustrated in FIG. **3**, whereby transmission gear **329** is formed as a combination gear that replaces transmission gear **323** and bevel gear **331**. Thus, gear pump **350** acts via the pumping action of transmission gears **322** and **329**, which run in tight tolerance against charge plate **328** and gear box housing cover **312**. This ready supply of pressurized hydraulic fluid can be drawn upon by an operator via actuation of valve **348**, here depicted as an electro-mechanical solenoid but understood to include manually actuated as well as other valve types. Should an operator not actuate valve **348** and permit fluid to accumulate in the accumulator **351** and charge gallery, a relief valve **352** is provided to allow fluid to be recycled to the intake zone of gear pump **350**. Gear pump **350** normally draws fluid from sump **360** via a vertical passage (not shown) that communicates with an opening in charge plate **328** at a first end of the passage, but fluid may also be drawn from outside sources such as an external reservoir (not shown) through inlet **371**. Inlet **371** is located proximate to the second, opposite end of the vertical passage (not shown) that feeds gear pump **350**. In either case, the fluid feeding pump **350** may be passed through an optional filter **353** located in the vertical passage (not shown). Use of inlet **371** requires that hydraulic fluid initially be circulated outside central gear box **310** through outlet **372**. By way of example, fluid may be circulated directly to an external oil cooler (not shown) or to the left and right side drive mechanisms **380** and **390** for cooling purposes before returning to inlet **371**. It should be understood that circulation of hydraulic fluid through the left and right side drive mechanisms **380** and **390** is only appropriate to those portions of the drive mechanisms that are similarly filled with hydraulic fluid. For example, if variable speed drives **381** and **391** are toroidal in nature, they will sealingly contain a traction oil isolated from their respective reduction gear sets **382** and **392**, which could contain hydraulic fluid. Thus, circulation of hydraulic fluid would only be appropriate to the portion of left and right side drive mechanisms **380** and **390** housing reduction gear sets **382** and **392**. Where variable speed drives **381** and **391** are hydraulic in nature, circulation of hydraulic fluid through the entire left and right side drive mechanisms **380** and **390** would be appropriate. Circulated hydraulic fluid may also be used to power a deck lift in ZT vehicles adapted for mowing. The primary purpose, however, is to provide a source of hydraulic fluid to actuate the hydraulic clutch and brake assembly **340** of the power take off mechanism. The specific workings of hydraulic power take off mechanisms, such as referenced in commonly owned U.S. Pat. No. 7,137,250, the disclosure of which is incorporated herein by reference, are well known in the art and shall only be described briefly herein.

The brake in hydraulic clutch and brake assembly **340** is biased toward engagement. Spring **343** acts on a piston **346** which in turn acts on a plurality of pins **345** (as best shown in FIG. **6**) to create frictional engagement of brake disks **364** which thereby hold cage **341** and power take off output shaft **342** static. Hydraulic fluid from gear pump **350** is routed to assembly **340** by opening valve **348**. The hydraulic fluid acts against the piston **346** which compresses spring **343** releasing the brake force on power take off output shaft **342**. The piston **346** further acts against clutch plates **347** to create frictional engagement therebetween which synchronizes the rotation of input shaft **321** with cage **341** and power take off output shaft **342**. When an operator closes valve **348**, cutting off the supply of pressurized hydraulic fluid to assembly **340**, hydraulic fluid is bled off to fluid sump **360**. In the absence of pressurized fluid, the piston can no longer overcome the spring force of spring **343**. As a result, the clutch plates **347** do not remain in frictional engagement and cage **341** (and the associated power take off output shaft **342**) is disengaged from input shaft **321**. Concurrently, spring **343** compresses the brake disks **364** into frictional engagement to stop rotation of cage **341** and the associated power take off output shaft **342**.

FIG. **3** further illustrates that the bevel gears **331** and **332** of distribution gear set **330** may be oriented in an opposite sense to reverse the rotation of central output shaft **333** from that previously observed in distribution gear sets **130** and **230** respectively. This variability creates flexibility in the manner in which left and right side drive mechanisms **380** and **390** may be applied. Jack shaft **324** is engaged to and drives bevel gear **331**. As shown in FIGS. **6** and **7**, bearing **327** rotationally supports jack shaft **324** in gear box housing cover **312**

FIGS. **4**, **5** and **6** depict a representative central gear box **310** in accordance with the principles of the invention outlined above in the description of the third embodiment of FIG. **3**, with the minor variation in FIG. **6** as previously described herein. Gear box housing **311** contains the central drive train and power take off mechanism. Gear box housing cover **312** is sealingly engaged to housing **311** by a plurality of fasteners **313** to form sump **360**. Housing cover **312** is formed with a plurality of bosses **314** by which central gear box **310** may be fastened to a vehicle frame. Housing cover **312** is also formed with a feature **312*a*** containing an internal volume which accommodates the charge gallery **354**, accumulator **351**, relief valve **352**, and clutch actuation valve **348**. Various features of the internal volume will require machining and consequently the passageways of the internal volume are sealed with plugs **316**. Charge outlet **372**, for example, is so sealed, as is the charge inlet **371** of housing **311**. Input shaft **321** is depicted as a hollow shaft with keyway, but it may take any necessary form known in the art to mate with an output shaft from the prime mover **305**. Central output shaft **333** is depicted as having the splined female configuration previously described.

FIG. **5** best illustrates the details of the internal volume of housing cover **312**, and the functionality of components disposed therein. Pressurized fluid from gear pump **350** enters charge gallery **354** through passage **361**. As fluid volume builds in charge gallery **354**, accumulator **351** reacts to accommodate the increasing volume. Accumulator **351** comprises a piston **351*a*** having an o-ring **351*b*** and spring **351*c***, wherein the piston **351*a*** is in sealing engagement with seat **354*b*** of charge gallery **354**. As fluid volume and pressure builds, piston **351*a*** is forced off seat **354*b*** and spring **351*c*** is compressed. An equilibration passage **363** is provided in the accumulator volume to permit any trapped air or fluid behind piston **351*a*** to be transferred to fluid sump **360**, allowing the piston **351*a*** to freely travel and preventing formation of a vacuum behind the piston **351*a***. Once piston **351*a*** has completed its full compression travel, increasing fluid volume and pressure will act on relief valve **352**, which is depicted as a ball **352*a*** and spring **352*b*** arrangement. It should be understood that other forms of relief valves known in the art would serve equally well and lie within the scope of the invention. Ball **352*a*** is in sealing engagement with seat **354*a*** of charge gallery **354**. As increased fluid volume and pressure overcome the spring force of spring **352*b***, ball **352*a*** is moved off seat **354*a***, where hydraulic fluid bypasses the ball **352*a*** and exits the charge gallery through relief passage **362** to the intake zone of gear pump **350** located in the volume above charge plate **328** directly below relief passage **362**. As fluid volume and pressure in the charge gallery decreases by cracking of relief valve **352**, or alternatively, actuation of valve **348**, ball **352*a*** returns to seat **354*a*** as the spring force of spring **352*b*** overcomes the opposing fluid force in charge gallery **354**.

Valve **348**, depicted herein as an electro-mechanical solenoid valve, is under operator control and permits pressurized hydraulic fluid from charge gallery **354** to actuate hydraulic clutch and brake assembly **340** when opened. The valve **348** comprises an electromagnet (not shown) and a valve body **348*a*** which is shown in sealing engagement with seat **354*c*** of charge gallery **354**. Actuation of valve **348** results in valve body **348*a*** being moved off seat **354*c***, permitting pressurized hydraulic fluid to flow through fluid passage **354*d*** and occupy the annular space **355** about input shaft **321**. As shown in FIGS. **5** and **6**, vertical passages **356*a-c*** adjacent input shaft **321** communicate with annular space **355** to provide pressurized hydraulic fluid to volume **315** below piston **346**, which is sealed by o-ring **376**. In the absence of pressurized fluid, clutch spring **343** acts against piston **346** to force a plurality of pins **345** into contact with piston **346**, thereby compressing brake disks **364** into frictional engagement with each other, arresting any rotation of cage **341** and power take off output shaft **342**. One set of the brake disks **364** is fixed to gear box housing **311**, and the intervening set of brake disks **364** is fixed to cage **341** by a gear form native to cage **341**. As pressurized fluid fills volume **315**, piston **346** acts against and overcomes the spring force of clutch spring **343**. Consequently, pins **345** no longer act to compress brake disks **364** into frictional engagement, permitting cage **341** and power take off output shaft **342** to rotate freely. As pressurized fluid continues to fill volume **315**, piston **346** begins to act against clutch plates **347**, compressing the clutch plates **347** into frictional engagement. The stack-up of clutch plates **347** consists of one set that is captured by a gear form **341*a*** native to the cage **341** and an interwoven set that is captured by a gear form native to transmission gear **322**, turning continuously therewith. The movement of the clutch plates are restricted vertically by retaining ring **344**, permitting frictional engagement to occur. As the engagement of the clutch plates **347** reaches maximum effect, with transmission gear **322** being splined to input shaft **321**, the rotation of input shaft **321** is imparted to cage **341** and power take off output shaft **342** which is splined thereto. The rotation of cage **341** is supported by bearing **365**, and the rotation of output shaft **342** is supported by bearing **366**. Hydraulic fluid is retained by a seal **367** and retaining ring **368** about output shaft **342**. To produce useful work, a pulley **349** may be affixed to output shaft **342** through means of a hub **369**, washer **373**, and hub nut **374**.

When valve **348** is closed, the flow of pressurized hydraulic fluid to the clutch is curtailed and the fluid accumulated in volume **315** returns to sump **360** through various spaces associated with component tolerances. As a result, the brake arrests movement of cage **341** and output shaft **342** as described above.

FIG. **7** provides a partial section view of the distribution gear set **330** which illustrates that space is available for distribution gear **332** to be mounted at either end of central output shaft **333** to achieve a particular rotational direction. Central output shaft **333** is rotationally supported by a bushing **378** at each end, and the gear box housing **311** is sealed at each end of central output shaft **333** by a lip seal **377**.

FIG. **8** illustrates a fourth embodiment of a modular drive system **400** which introduces a dedicated charge pump **459** to provide a source of pressurized hydraulic fluid for the hydraulic clutch and brake assembly **440** of the power take off mechanism. Charge pump **459** may be of any known design such as a gerotor or geroller pump. The use of such a charge pump eases manufacturing tolerances for the transmissions gear set **420** as compared to that previously described for the transmission gear set **320** of the third embodiment. Jack shaft **424** extends through transmission gear **423** to drive charge pump **459** in addition to driving distribution gear set **430**. As previously described for gear pump **350**, charge pump **459** draws hydraulic fluid from fluid sump **460** through optional filter **453**. Pressurized hydraulic fluid is discharged to a charge gallery **454** (as best shown in the FIG. **10**) in fluid communication with an accumulator **451**, a relief valve **452** and hydraulic clutch actuation valve **448**. The workings of the hydraulic clutch and brake assembly **440** and the related hydraulic circuit are as previously described for the third embodiment. It should also be recognized that charge pump **459**, with its greater efficiency, is more suited to providing hydraulic fluid for external circulation via outlet **472** and inlet **471** than gear pump **350**.

FIGS. **9**, **10** and **11** depict a representative central gear box **410** in accordance with the principles of the invention outlined above in the description of the fourth embodiment, as shown in FIG. **8**. Similar to central gear box **310**, FIG. **11** illustrates a compact variation on the central drive train illustrated in FIG. **8**, whereby transmission gear **429** is a combination gear that replaces transmission gear **423** and bevel gear **431**. The differences in the structure and function of central gear box **410** as compared to that just described for central gear box **310** will now be addressed. Central gear box housing **411** and housing cover **412** have been altered to accommodate a charge pump **459** and the requisite fluid inlet passage **418** and fluid discharge passage **417**. These requisite passages predominately occupy feature **412***a* of housing cover **412**. Charge pump **459** is depicted herein as a gerotor pump, but other known pump types such as a geroller pump are contemplated by the invention. Notably, inlet **471** has been moved to the perimeter of housing **411** away from any potential interference with pulley **449** in recognition of the capacity of charge pump **459** to do more than provide fluid for hydraulic clutch and brake assembly **440**. Outlet **472** can be utilized to circulate pressurized hydraulic fluid outside central gear box **410** for various purposes prior to intake at inlet **471**.

Charge pump **459** draws hydraulic fluid from either fluid sump **460** or inlet **471** through a series of passages beginning with vertical inlet passage **479**. Vertical inlet passage **479** consists of an upper portion formed in gear box housing cover **412** and a lower portion formed in gear box housing **411**. The joint between the upper and lower portions may be sealed with an o-ring (not shown), or a tube insert with an o-ring at both ends of the tube (not shown). Hydraulic fluid then passes through inlet passage **418** which communicates with the arcuate port **419***a* to feed charge pump **459**. The inner rotor **496** is fixed to jack shaft **424** and works in combination with outer rotor **497** to pump hydraulic fluid out arcuate port **419***b* and into fluid discharge passage **417**. Hydraulic fluid then exits discharge passage **417** at opening **461** to fill charge gallery **454**. Inner rotor **496** and outer rotor **497** utilize charge plate **498** as a running surface. Charge plate **498** is mounted to housing cover **412** using fasteners **486** and features a fluid annulus **488** to provide lubrication to the gears of the central drive train. The additional loading imparted to jack shaft **424** by charge pump **459** is mediated by the addition of thrust washer **487**.

The operation of the common elements of central gear box **410**, here labeled as **400** series elements, are as described for those same elements of central gear box **310** and will not be further discussed. One distinction should be noted in the operation of relief valve **452**, which here exhausts directly to sump **460** in the absence of an equivalent to charge plate **328** directly below the gears of the central drive train.

FIG. **12** illustrates a fifth embodiment of a modular drive system **500** which differs from the fourth embodiment in two primary ways, namely, the addition of an auxiliary pump **558** and a new configuration for the distribution gear set **530**. Prime mover **505** provides motive force to input shaft **521** which powers transmission gear set **520**. More specifically, transmission gear **522**, which is engaged to input shaft **521**, powers both transmission gears **523** and **525** in the same rotational sense. Transmission gear **523** provides motive force to both charge pump **559** and distribution gear set **530** via jack shaft **524**. Transmission gear **525** provides motive force to auxiliary pump **558** via auxiliary shaft **526**. Auxiliary pump **558** may be of various types, including but not limited to, gear pumps, gerotor pumps or geroller pumps. Auxiliary pump **558** draws hydraulic fluid from fluid sump **560** to make up for any lost volume in the external loop between auxiliary outlet **502** and auxiliary inlet **501**, wherein the fluid drawn from fluid sump **560** may be passed through optional filter **553** prior to intake. The pressurized fluid discharged from auxiliary pump **558** exits central gear box **510** at auxiliary outlet **502** and returns through either dedicated auxiliary inlet **501** or inlet **571** which facilitates filtering. In addition to the external uses described for pressurized fluid from gear pump **350** and charge pump **459**, the pressurized fluid from auxiliary pump **558** can be used to power auxiliary devices such as trimmers or augers attached to utility vehicles. An auxiliary relief valve **557** is included in the auxiliary pump's hydraulic circuit to relieve fluid that is not being utilized by an external device. It should also be noted that the charge relief valve **552** provided for the fifth embodiment differs from charge relief valve **452** in that it exhausts to the intake of charge pump **559**, as opposed to discharging directly to sump **560**.

Distribution gear set **530** illustrates a further drive train variant intended to permit the input shafts **583** and **593** to the left and right side drive assemblies, **580** and **590** respectively, to be driven in opposite rotational directions. The purpose is to allow variable drive mechanisms **581** and **591** to be identical units, not mirror-image units; thus, providing a cost advantage during manufacture. Jack shaft **524** is engaged to and drives bevel gear **531** which drives corresponding left and right side bevel gears **532***a* and **532***b* respectively. The latter gears are each engaged to and drive a corresponding central output shaft **533***a* and **533***b* respectively. Each central output shaft is thereafter engaged to a corresponding input shaft **583** or **593** for the left and right side drive mechanisms, providing motive force thereto to ultimately propel wheels **589** and **599**.

FIG. **13** illustrates a sixth embodiment of a modular drive system **600** which also uses a dedicated charge pump **659** to provide a source of pressurized hydraulic fluid for a dual hydraulic clutch and brake assembly **640**, where both clutches are disposed on the input shaft **621** to selectively engage the central drive train and the power take off mechanism respectively. Many of the elements in this embodiment that may be substantially identical to those previously described and which differ only in the initial numeral in this description will not be described in detail herein.

FIGS. **14**, **15**, **16** and **17** depict a representative central gear box **610** in accordance with the principles of the invention outlined above in the description of the sixth embodiment, as shown in FIG. **13**. Similar to central gear boxes **110**, **210**, **310** and **410**, FIG. **16** illustrates a compact variation on the central drive train illustrated in FIG. **13**, whereby transmission gear **629** is a combination gear that replaces transmission gear **623** and bevel gear **631**. FIG. **17** provides a further view of transmission gear **629** and its integration with distribution gear **632** and central output shaft **633**. The form and function of the distribution gear set **630** is as described for the third embodiment in FIG. **7** and will not be further detailed herein. The differences in the structure and function of central gear box **610** as compared to that previously described for central gear box **410** will now be addressed. Central gear box housing **611** and housing cover **612** have been altered to accommodate certain changes in this embodiment, such as the use of the dual hydraulic clutch and brake assembly **640**, a pair of hydraulic clutch actuation valves **648***a* and **648***b* to individually control the two clutch mechanisms, a charge pump **659** on the input shaft **621** and the corresponding charge gallery **654** that provides hydraulic fluid to both clutch mechanisms.

Charge pump **659** is driven directly by input shaft **621** and pressurized hydraulic fluid is discharged to a charge gallery **654** in fluid communication with an accumulator **651**, a relief valve **652** and a pair of hydraulic clutch actuation valves **648***a* and **648***b*. Charge pump **659** draws hydraulic fluid from either fluid sump **660** or inlet **671** through a series of passages beginning with vertical inlet passage **679**. Hydraulic fluid then passes through inlet passage **618** which communicates with the arcuate port **619***a* to feed charge pump **659**. The inner rotor **696** is fixed to input shaft **621** and works in combination with outer rotor **697** to pump hydraulic fluid out arcuate port **619***b* and into charge gallery **654** through the opening **661** formed by the intersection of arcuate port **619***b* and charge gallery **654**. Thus, pressurized hydraulic fluid is made available to charge accumulator **651** and selectively actuate the dual clutches upon demand. A vertical passage **603** leading from charge gallery **654** to the hydraulic clutch actuation valves **648***a* and **648***b* provides the necessary fluid communication. Charge relief valve **652**, whose ball and spring are now vertically oriented below charge gallery **654**, exhausts fluid directly to sump **660** via a vertical passage (not shown) through charge plate **698**, and not to an intake passage for the charge pump **659**.

This embodiment provides a dual clutch and brake assembly **640** having two sets of clutch plates **647***a*, **647***b* and a pair of corresponding hydraulic clutch actuation valves **648***a*, **648***b*. The addition of a clutch mechanism to selectively engage the drive train reduces wear on drive train components and maximizes power transmission to the power take off mechanism when the power take off mechanism is operated with the vehicle stationary, such as when powering an auger for digging post holes. Both of the clutches share a common clutch plate carrier **609** that is fixed to and rotates with input shaft **621**. Half of each set of the clutch plates (**647***a* for the drive train clutch and **647***b* for the power take off clutch) are fixed to the common clutch plate carrier **609** by corresponding gear forms and rotate therewith. It should be understood that the specific shape of the interface between input shaft **621** and the clutch plates **647***a* and **647***b* is not critical provided they interlock and rotate together. Transmission gear **622** is formed with a cage portion **622***a* of the drive train clutch which similarly captures the clutch plates **647***a* that are alternately interlaced with those of input shaft **621**. Retaining ring **644***a* maintains the stack-up and permits frictional engagement of the drive train clutch plates **647***a*. Cage **641***b* of the power take off clutch captures the clutch plates **647***b* that are alternately interlaced with those of input shaft **621**. Retaining ring **644***b* maintains the stack-up and permits frictional engagement of power take off clutch plates **647***b*.

The form and function of both these hydraulic clutches is similar to that previously described for the third and fourth embodiments. The drive train clutch and the power take off clutch each have a clutch spring (**643***a* and **643***b* respectively) that acts against a piston (**646***a* and **646***b* respectively) to remove compressive force from the clutch plates (**647***a* and **647***b* respectively) and bias the clutches to a disengaged state. In the case of the power take off clutch, the action of clutch spring **643***b* further brings piston **646***b* into engagement with a plurality of pins **645** which in turn bring the stack-up of brake plates **664** into frictional engagement, thereby arresting any rotation of cage **641***b* and the affixed power take off output shaft **642**. Brake plates **664** are alternately retained in their stack-up by gear box housing **611** and cage **641***b*. The drive train clutch and the power take off clutch are each brought into frictional engagement when a hydraulic clutch actuation valve (**648***a* and **648***b* respectively) is opened to permit pressurized hydraulic fluid in the vertical passage **603** to fill an annular space (**655***a* and **655***b* respectively) about the cage (**622***a* and **641***b* respectively) and then traverse a passage (**656***a* and **656***b* respectively) to fill the volume (**615***a* and **615***b* respectively) adjacent the piston (**646***a* and **646***b* respectively) and opposite the clutch spring (**643***a* and **643***b* respectively). A plurality of o-rings **608** helps to retain fluid in the annular spaces **655***a* and **655***b*. The pressurized hydraulic fluid acts against the piston (**646***a* and **646***b* respectively) and the opposing clutch spring forces to bring the clutch plates (**647***a* and **647***b* respectively) into frictional engagement, their movement arrested against the retaining ring (**644***a* and **644***b* respectively). Thus, transmission gear **622** is brought into rotational synchronization with input shaft **621** to power transmission gear **629** and ultimately central output shaft **633**. The movement of piston

646*b* against the power take off clutch plates **647***b* correspondingly removed the transmitted spring forces from the plurality of pins **645** and brake plates **664**. Thus, cage **641***b* and the affixed power take off output shaft **642** become rotationally free to synchronize with the rotation of input shaft **621** as the frictional engagement of the power take off clutch plates **647***b* comes into full effect. The clutches are disengaged when the hydraulic clutch actuation valves **648***a* and **648***b* are closed, removing the hydraulic force from pistons **646***a* and **646***b*, which then come under the influence of the clutch springs **643***a* and **643***b*, taking the clutch plates **647***a* and **647***b* out of frictional engagement. Excess hydraulic fluid from volumes **615***a* and **615***b* migrates to sump **660** through the various tolerances between component parts as pistons **646***a* and **646***b* move under the influence of clutch springs **643***a* and **643***b*.

Rotational support for the various shafts and cages of central gear box **610** and hydraulic seals for the various components are similar to the scheme previously described for the third and fourth embodiments and will not be further addressed herein. As before, a pulley **649** may be fastened to power take off output shaft **642** with a hub **669**, washer **673** and hub nut **674** to perform useful work.

FIG. **18** illustrates a seventh embodiment of a modular drive system **700** where a hydraulic clutch assembly **740***a* is disposed on jack shaft **724** to selectively drive distribution gear set **730**, while a hydraulic power take off clutch and brake assembly **740***b* is disposed on input shaft **721** in a manner similar to that shown in FIG. **3**. A gear pump **750** comprising transmission gears **722** and **723** is driven by input shaft **721** and provides pressurized hydraulic fluid to a charge gallery **754** (as best shown in FIG. **20**) that serves to supply hydraulic fluid to actuate clutch assembly **740***a* and clutch and brake assembly **740***b*. A pair of clutch actuation valves **748***a* and **748***b* permit an operator to individually control both clutch assemblies. An intermediate shaft **704** is engaged to and selectively driven by cage **741***a* upon actuation of clutch **740***a*. Intermediate shaft **704** is further engaged to and drives bevel gear **731** which drives corresponding bevel gear **732**. The latter gear is engaged to and drives central output shaft **733** to power left and right side drive mechanisms **780** and **790**.

FIGS. **19**, **20** and **21** depict a representative central gear box **710** in accordance with the principles of the invention outlined above in the description of the embodiment of FIG. **18**. Gear box housing **711** contains the central drive train and power take off mechanism. Representative central gear box **710** varies slightly from the depiction of the seventh embodiment in FIG. **18** in that cage **741***a*, intermediate shaft **704**, and bevel gear **731** are combined as a single element, combination gear/cage **770** to conserve space and material.

FIG. **20** best illustrates the details of the internal volume of housing cover **712**, and the functionality of components disposed therein. Housing cover **712** is also formed with a feature **712***a* containing an internal volume which accommodates the charge gallery **754**, relief valve **752**, both clutch actuation valves **748***a* and **748***b* and corresponding fluid passages **754***a* and **754***b* to accommodate fluid flow to the respective clutches **740***a*, **740***b*. As described for the gear pump **350** of central gear box **310**, gear pump **750** acts via the pumping action of transmission gears **722** and **723**, which run in tight tolerance against a charge plate **728** and gear box housing cover **712**. Pressurized fluid from gear pump **750** enters charge gallery **754** through passage **761**.

Valves **748***a*, **748***b* are depicted herein as electro-mechanical solenoid valves under operator control and are similar in construction to the valves described in connection with other embodiments. Valves **748***a*, **748***b* permit pressurized hydraulic fluid from charge gallery **754** to actuate respective hydraulic clutch assemblies **740***a* and **740***b* when opened.

The operation of the clutch and brake assembly **740***b* will be discussed first; this assembly is similar in operation to the equivalent assembly **340** discussed above. Actuation of valve **748***b* permits pressurized hydraulic fluid to occupy the annular space **755***b* about input shaft **721**. As shown in FIGS. **20** and **21**, vertical passages **756***a-c* adjacent input shaft **721** communicate with annular space **755***b* to provide pressurized hydraulic fluid to volume **715***b* below piston **746***b*. In the absence of pressurized fluid, clutch spring **743***b* acts against piston **746***b* which is forced into contact with a plurality of pins **745**, which thereby compress brake disks **764** into frictional engagement with each other, arresting any rotation of cage **741***b* and the affixed power take off output shaft **742**. One set of the brake disks **764** is fixed to gear box housing **711**, and the intervening set of brake disks **764** is fixed to cage **741***b* by a gear form native to cage **741***b*. As pressurized fluid fills volume **715***b*, piston **746***b* acts against and overcomes the spring force of clutch spring **743***b*. Consequently, pins **745** no longer act to compress brake disks **764** into frictional engagement, permitting cage **741***b* and power take off output shaft **742** to rotate freely. As pressurized fluid continues to fill volume **715***b*, piston **746***b* begins to act against clutch plates **747***b*, compressing the clutch plates **747***b* into frictional engagement. One set of the clutch plates **747***b* is captured by a gear form native to cage **741**, while the intervening set of clutch plates **747***b* is captured by a gear form native to transmission gear **722**. The movement of the clutch plates **747***b* are restricted vertically by retaining ring **744***b*, permitting frictional engagement to occur. As the engagement of the clutch plates **747***b* reaches maximum effect, with transmission gear **722** being splined to input shaft **721**, the rotation of input shaft **721** is imparted to cage **741***b* and power take off output shaft **742** which is splined thereto. The rotation of cage **741***b* is supported by bearing **765**, and the rotation of output shaft **742** is supported by bearing **766**. Hydraulic fluid is retained by a seal **767** and retaining ring **768** about output shaft **742**. To produce useful work, a pulley **749** may be affixed to output shaft **742** through means of a hub **769**, washer **773**, and hub nut **774**. When valve **748***b* is closed, the flow of pressurized hydraulic fluid to the clutch is curtailed and the fluid accumulated in volume **715***b* returns to sump **760** through various spaces associated with component tolerances. As a result, the brake arrests movement of cage **741***b* and output shaft **742** as described above.

Next, a description will be given of clutch assembly **740***a* mounted on jack shaft **724** and controlled by clutch actuation valve **748***a*. Clutch assembly **740***a* operates as described for the clutch portion of clutch and brake assembly **740***b* with two exceptions. First, when clutch actuation valve **748***a* is opened, pressurized hydraulic fluid flows into passage **754***a* and then through the support bearing **727** for the jack shaft **724** before entering annular space **755***a*. (Jack shaft **724** is rotationally supported at a second end by a bushing or bearing **707**.) The hydraulic fluid is then distributed to the volume **715***a* below piston **746***a* by way of three vertical passages **756***d* (one of three shown) adjacent jack shaft **724**. The flow of pressurized hydraulic fluid to volume **715***a* counteracts the spring force of clutch spring **743***a* to force piston **746***a* against a plurality of clutch plates **747***a*. The frictional engagement of the clutch plates **747***a* thereby attained places combination gear/cage **770** into synchronous rotation with transmission gear **723** and jack shaft **724**.

Bevel gear **732**, which is rotationally mated to combination gear/cage **770**, then drives central output shaft **733**. The second difference between clutch assembly **740***a* and clutch and brake assembly **740***b* is realized by the absence of a brake on the output of clutch assembly **740***a*, wherein clutch spring **743***a* merely biases the clutch to a non-engaged state as its action on piston **746***a* does not result in frictional engagement of brake disks via pin displacement. When valve **748***a* is closed, the flow of pressurized hydraulic fluid to the clutch is curtailed and the fluid accumulated in volume **715***a* returns to sump **760** through various spaces associated with component tolerances. As a result, the powered rotation of central output shaft **733** is ceased.

FIGS. **22-27** illustrate another central gear box **810** in accordance with the principles of the invention; most similar to, but differing from the central gear box **410** of FIG. **8**. The major differences comprise a bolt-on charge pump **859** at the end of jack shaft **824** opposite housing cover **812**, rather than a charge pump set within the housing cover **812**. The vertical input shaft **821** is a though-shaft design that allows the central gear box **810** to be mounted with either end of the through-shaft oriented in an upward direction. This permits flexibility in the height at which pulley **849** is utilized. The differences further comprise an electro-mechanical valve **848**, illustrated here as a solenoid valve, normally open to sump **860** to discharge accumulated hydraulic fluid from the charge gallery **854**, the accumulator **851**, and the brake and clutch assembly **840**. Electro-mechanical valve **848** is closed to actuate the clutch. Additionally, the charge gallery **854** is internal to the gear box housing **811**, and not a formed feature of the housing cover **812**. Central gear box **810** also has a distribution gear set **830** similar to the teachings of FIG. **12**, wherein two oppositely rotating central output shafts **833***a* and **833***b* are present. Lastly, central gear box **810** comprises a screw-on filter **853** and associated passageways to remove debris from the hydraulic fluid of sump **860**. The details of central gear box **810** will be discussed below.

FIG. **22** shows central gear box **810** in a first orientation with pulley **849** below the unit. Gear box housing cover **812** is sealingly engaged to gear box housing **811** by a plurality of fasteners **813** to form sump **860**. Housing cover **812** is formed with a plurality of bosses **814** by which central gear box **810** may be fastened to a vehicle frame. The width of charge pump housing **898** and the lower portion of gear box housing **811** to which it is fastened by a pair of fasteners **886** (as best shown in FIG. **23**) is narrow enough to permit a v-belt (not shown) utilized by pulley **849** to freely pass adjacent to housing **811**. Such a belt provides motive force to an implement such as a mowing deck (not shown) when the power take off mechanism is engaged. Screw-on filter **853**, which threads onto the intake port **818** for charge pump **859** extending through an opening **811***c* in the side of gear box housing **811**, is readily accessible for service by an operator. The screw-on filter **853** seals to housing **811** in a conventional manner. An expansion tube **872** threads to a port **811***b* on gear box housing **811** (as best shown in FIG. **26**) that permits a hose (not shown) to place sump **860** in fluid communication with an expansion tank (not shown). Expansion tube **872** may alternatively be threaded to the opposite side of gear box housing **811** at port **811***a* (shown sealed with a plug **816** in FIG. **23**).

FIG. **23** shows central gear box **810** in a second, inverted orientation with pulley **849** placed above the unit and the other end of through-shaft **821** clearly visible. A circular retaining plate **873** is secured by fasteners **874** to the cage **841** of the brake and clutch assembly **840**, affixing the pulley **849** to the cage **841** and retaining a shaft seal **875***b* (as best shown in FIG. **25**).

FIGS. **25** and **26** are useful in describing the functional differences between gear box **810** and previous embodiments. Input shaft **821** is driven by the output shaft of a prime mover (not shown), and in turn powers transmission gear **822**, which is fixed to input shaft **821** by corresponding gear forms. Input shaft **821** is rotationally supported by bearing **804**, shaft seal **875***a* seals the opening for input shaft **821** in housing cover **812**, and thrust washer **887***a* supports input shaft **821**. Transmission gear **822** drives combination gear **829**, which is splined to jack shaft **824** at its proximal end. Jack shaft **824** is rotationally supported by bearing **827**. The rotation imparted to jack shaft **824** is transferred to the inner rotor **896** of gerotor style charge pump **859** to which jack shaft **824** is engaged at its distal end. The pumping action of inner rotor **896** and corresponding outer rotor **897** draws hydraulic fluid from sump **860** through opening **811***c* and into screw-on filter **853**. After passing through the various filter elements (not shown) of screw-on filter **853**, hydraulic fluid enters intake port **818** and is finally drawn into charge pump **859** via kidney-shaped port **819***a*. The running surface **811***d* for inner rotor **896** and outer rotor **897** is integral to an exterior face of gear box housing **811**. Pressurized hydraulic fluid exits charge pump **859** through the corresponding kidney shaped exhaust port **819***b*, and thereafter travels to a charge gallery **854** in fluid communication with an accumulator **851**, a relief valve **852**, and a clutch actuation valve **848**.

Valve body **848***a* is normally unseated or open, permitting hydraulic fluid from the charge gallery **854** and accumulator **851** to be recirculated by charge pump **859** through sump **860** and ultimately, filter **853**, wherein fine metal debris from the gear train or other contaminants may be trapped to extend the service life of key components. When valve **848** is actuated, valve body **848***a* is seated against the periphery of charge gallery **854**, closing off the recirculation of hydraulic fluid back to sump **860**. Instead, pressurized hydraulic fluid begins to accumulate in charge gallery **854**, accumulator **851**, and the clutch assembly **840**. Accumulator **851** acts as a source of backup power for clutch assembly **840** and also dampens the action of clutch assembly **840**, reducing wear on its components and those mechanically driven by the power take off assembly. Accumulator **851** consists of a piston **851***a* having a tight tolerance with the accumulator bore and a spring **851***c* retained by a plug **816**. The accumulator has an equilibration passage **863** for entrained air or hydraulic fluid to prevent the piston from seizing.

Brake and clutch assembly **840** has features similar to previous embodiments described herein, including an annular passageway **855** in cage **841** that is fed by the charge gallery **854** at a narrowed portion **854***d*. As shown in FIG. **25**, a plug **806** seals off the end of charge gallery **854** opposite narrowed portion **854***d*, the end being an artifact of the machining operations necessary to form charge gallery **854** and the narrowed portion **854***d*. Vertical passageway **856** permits hydraulic fluid from annular passageway **855** to enter the volume **815** below piston **846**. O-ring **876** prevents hydraulic fluid bypassing piston **846**. Any hydraulic fluid bypassing the o-ring **808** below annular passage **855** enters passageways **803***a* and **803***b* of cage **841**, whereby lubrication is provided to the interface between input shaft **821** and cage **841**. Seal **867** seals the opening for cage **841** in gear box housing **811**. The operation of the brake and clutch assembly **840** is similar to the operation of its counterparts in the previously described third, fourth, sixth and seventh embodiments and will not be described in detail.

The spring force of clutch spring **843**_a_, which is retained by a washer **843**_b_ and retaining ring **843**_c_ about cage **841**, works against piston **846** and forces a plurality of pins **845** to act on brake disks **864**, bringing the brake disks **864** into frictional engagement and arresting any rotational movement of cage **841** and the pulley **849** engaged thereto. Thrust washers **887**_a_ and **887**_b_ support any axial movement of cage **841**. The accumulation of pressurized hydraulic fluid in volume **815** disengages the brake and engages the clutch of assembly **840**. The spring force of clutch spring **843**_a_ is overcome by the opposing fluid force exerted on piston **846**, whereby the plurality of pins **845** no longer hold the brake disks **864** in frictional engagement. Cage **841** and the engaged pulley **849** are now free to rotate in synchronization with input shaft **821** as the clutch plates **847** come into frictional engagement by action of the fluid force applied through piston **846**. Retaining ring **844** serves as a stop for the clutch plates **847** to enable their engagement.

As fluid pressure continues to build with the clutch engaged, charge relief valve **852** will crack at a predetermined pressure to protect the various seals and o-rings associated with the input shaft **821** and cage **841**. While shown as a simple ball **852**_a_ and spring **852**_b_ valve retained by a plug **852**_c_ having an integral relief passage **862**, relief valve **852** can be of any known design in the art, such as a poppet valve.

As outlined herein, rotational support for the various shafts and cages of central gear box **810** and hydraulic seals and o-rings for the various components are similar to the scheme previously described for the third and fourth embodiments.

FIGS. **25** and **27** best illustrate the distribution gear set **830**. Combination gear **829** is engaged to and driven by the rotation of transmission gear **822**, which is splined to input shaft **821**. Combination gear **829** has a secondary gear form **829**_a_ that engages and drives a pair of distribution gears **832**_a_ and **832**_b_. Though depicted as spiral bevel gears with the attendant benefits of noise reduction and higher speed capacity, secondary gear form **829**_a_ and distribution gears **832**_a_ and **832**_b_ could also be generic bevel gears. Each of the distribution gears **832**_a_ and **832**_b_ are splined to independent output shafts **833**_a_ and **833**_b_ respectively. The output shafts **833**_a_ and **833**_b_ are rotationally supported at a first end by a bearing **878** retained in a pocket of the gear box housing **811** (retained therein by a retaining ring **868**), and rotationally supported at a second end by cradles **811**_e_ formed in the gear box housing **811**. Both output shafts **833**_a_ and **833**_b_ are retained axially by retaining rings **833**_c_ to prevent contact with jack shaft **824**, and are sealed by a shaft seals **877** at their exterior ends. As noted previously, the output shafts **833**_a_ and **833**_b_ rotate in opposite directions, permitting the use of identical left and right side drive mechanisms.

FIG. **28** depicts the underside of a ZT vehicle **1000** integrating the central gear box **810** of FIGS. **22-27** and left and right side transaxles **1010**_a_ and **1010**_b_, in this instance hydrostatic transaxles, which will be described in more detail below. It should be noted that transaxles **1010**_a_ and **1010**_b_ are just one example of “plug and play” drive mechanisms that integrate with central gear box **810** to form a modular drive system. Vehicle **1000** comprises engine/prime mover **1005** mounted on vehicle frame **1002**. Deck drive belt **1006** couples the output pulley **849** of the power take off mechanism with mowing deck **1004**. Front wheels **1009**, typically caster wheels, are attached to vehicle frame **1002** in front of mowing deck **1004**. Rear wheels **1008** are mounted to the left and right side transaxles **1010**_a_ and **1010**_b_ behind mowing deck **1004**. In addition, cross brace **1012** is mounted between the housings of transaxles **1010**_a_ and **1010**_b_ to provide additional rigidity and limit driveline flexion.

FIG. **29** shows output shaft **1007** of engine/prime mover **1005** extending through vehicle frame **1002** to mate with the input shaft **821** (not shown) of gear box **810** and further shows input shafts **1028**_a_ and **1028**_b_ extending out of transaxles **1010**_a_ and **1010**_b_, respectively. Return to neutral assemblies **1048**_a_ and **1048**_b_ are shown mounted on transaxles **1010**_a_ and **1010**_b_ and extending though vehicle frame **1002** for protection from ground debris.

FIGS. **30** and **31** show a transaxle **1010**_c_ having a central housing **1020** with pump housing cover **1018** and gear housing cover **1024** fastened thereto by fasteners **1026**. Transaxle **1010**_c_ varies from the previously depicted transaxles **1010**_a_ and **1010**_b_, in that its return to neutral mechanism **1048** and control arm **1050** are rotated one hundred-eighty (180) degrees from those of transaxles **1010**_a_ and **1010**_b_ to accommodate a particular vehicle's control layout. It is understood that the length of input shaft **1028**, depicted as shorter in transaxle **1010**_c_, can vary to accommodate a host of vehicle frame widths. Central housing **1020** has a plurality of bosses **1021** by which transaxle **1010**_c_ may be fastened to vehicle frame **1002** or cross brace **1012**. Brake arm **1084** is retained on brake/cam shaft **1086** by clip **1088**. Rotation of brake arm **1084** rotates a brake/cam shaft **1086** to bring an internal, frictional element (not shown) into frictional engagement with a rotating ring gear **1070** (as shown in FIG. **32**). Such a brake mechanism is suitable as a parking brake. Central housing **1020** also has an expansion tank port **1022** which allows connection to such an external expansion tank (not shown). Bypass rod with cam **1080** extends into central housing **1020** adjacent to pump housing cover **1018**. Return to neutral assembly **1048**, which acts on control arm **1050**, and consequently trunnion arm **1052** (as best seen in FIG. **32**), is mounted in part on trunnion arm **1052** and pump housing cover **1018**. The workings of a scissor-arm return to neutral mechanism, as depicted herein, is illustrated in commonly-owned U.S. Pat. No. 6,487,857, incorporated by reference herein, and will not be discussed further. Pump input shaft **1028** has male splines **1029** with a rounded or tapered profile to provide a flexible joint that negates the effect of vehicle frame flexion.

FIG. **32** shows the internal components of transaxle **1010**_c_ comprising input shaft **1028**, pump assembly **1032**, center section **1040**, motor assembly **1036**, and motor output shaft **1046** arranged in-line. The rotation of input shaft **1028** is supported by bearing **1060**. Center section **1040** is attached to central housing **1020** by fasteners **1044** and has fluid passages **1042** to allow circulation of hydraulic fluid in a closed loop from pump assembly **1032** to motor assembly **1036**. A pair of check plugs **1082** and associated check balls (not shown) are disposed in fluid passages (not shown) in center section **1040** to permit fluid exchange between a sump **1061** and fluid passages **1042**. The check plugs **1082** and bypass rod with cam **1080** act in concert to effect a bypass of the closed loop by opening the otherwise closed hydraulic circuit to a sump **1061**. Rotation of bypass rod with cam **1080** brings the cam into displacing contact with the check balls to open the hydraulic circuit.

For adjusting the output of transaxle **1010**_c_, there is swash plate **1054** in contact with pump thrust bearing **1056** against which pump pistons **1033** travel. The direction of rotation of pump assembly **1032** is fixed by the direction of rotation of input shaft **1028**, to which pump assembly **1032** is nonrotatably joined. As will be understood by those of ordinary skill in the art, swash plate **1054** may be moved to a variety of positions to vary the displacement of pump pistons **1033**, the corresponding rotational speed and direction of rotation of motor assembly **1036**, and the corresponding output of motor shaft **1046**. Motor pistons **1037** move against thrust bearing **1058** which is set at a fixed, non-neutral angle of displacement. Movement of the swash plate **1054** is accomplished by operator controlled movement of control arm **1050** via a control linkage (not shown) fastened to the control arm **1050**. Rotation of control arm **1050** causes a corresponding rotation of trunnion arm **1052**, fastened thereto, which is engaged to swash plate **1054**. Thus, swash plate **1054** may be swung fore and aft through an arc to effect displacement of pump pistons **1033**.

Generally, as the angle of swash plate **1054** is varied in one direction from the neutral position, the stroke of the pump pistons **1033** is varied, which then drives the motor assembly **1036** at a speed determined by the volume of the fluid displaced by the pump pistons **1033**. As the angle of the swash plate **1054** is decreased to pass through the neutral position, the direction of rotation of motor assembly **1036** and its corresponding output motor shaft **1046** is reversed. The speed of the motor is again determined by the volume of fluid displaced by the pump pistons **1033**.

As best shown in FIG. **32**, in this illustrated embodiment, axle shaft **1062** includes a proximal end **1062***a*, a distal end **1062***b*, a body extending from proximal end **1062***a* to distal end **1062***b*, and a threaded section **1062***c* adjacent distal end **1062***b*. Axle shaft **1062** is rotationally supported by inner axle bearings **1064** and outer axle bearings **1066** such that proximal end **1062***a* is disposed inside transaxle **1010***c* and distal end **1062***b* is disposed external to transaxle **1010***c*. The body of axle shaft **1062** includes a first spline **1062***d* extending from distal end **1062***b* to a location along the body adjacent to transaxle **1010***c* and a second spline **1062***e* located on a portion of the body disposed inside transaxle **1010***c*. To drive axle shaft **1062**, reduction gearing is provided to couple axle shaft **1062** to motor shaft **1046**, imparting thereto an appropriate speed and torque. Thus, motor shaft **1046** is splined to pinion gear **1068** which drives ring gear **1070**. Ring gear **1070** in turn drives sun gear **1072** which drives planet gears **1074**. Because the planetary ring gear **1078** is held stationary, the planet gears **1074** drive planet gear carrier **1076** which is splined to axle shaft **1062** (and, in particular, to second spline **1062***e*). Other known reduction schemes may be employed to impart an appropriate speed and torque to axle shaft **1062**.

FIGS. **33-36** depict another embodiment of a hydrostatic transaxle (shown as transaxles **1110***a*, **1110***b* and **1110***c*) suitable to integrate with central gear box **810** and form a modular drive system. The use of central gear box **810** is understood to be that of an exemplar central gear box in accordance with the principles of the invention. Transaxle **1110***c* is similar to the embodiment shown in FIGS. **28-32** in every major respect but two; first, in this embodiment the pump and motor assemblies (**1132** and **1136** respectively) are arranged in a parallel configuration with the motor shaft **1146** offset from pump shaft **1128**, and second, main housing **1116** replaces central housing **1020** and pump housing cover **1018**. This transaxle embodiment also differs in five minor respects. First, center section **1140** is wider than center section **1040** to accommodate the adjacent running surfaces (not shown) for motor assembly **1135** from pump assembly **1132**, which correspondingly creates fluid passages **1142** that are longer than fluid passages **1042**. Second, fasteners **1144** are inverted as compared to fastener **1044**. Third, pinion gear **1168** is located below sun gear **1172**, whereas in the previous embodiment pinion gear **1068** was above sun gear **1072**. Fourth, there is no expansion tank port in this embodiment, though one could be added. Fifth and finally, transaxle **1110***c* has a narrower and taller profile as compared to transaxle **1010***c*. Many of the elements in this embodiment are substantially identical to those previously described and will not be described herein.

FIG. **37** depicts modular drive system **1201** comprising central gear box **110** and left and right side drive mechanisms, transaxles **1210***a* and **1210***b* respectively, which in this instance are toroidal drives. The individual components of modular drive system **1201** are mounted to vehicle frame **1202** by a variety of fasteners **1293**, **1294** and **1295**. In addition, a cross brace **1212** is mounted between the housings of transaxles **1210***a* and **1210***b* to provide additional rigidity and limit driveline flexion. As previously mentioned, the nature of the joint between the central output shafts **133***a* and **133***b* of central gear box **110** and the input shafts **1228***a* and **1228***b* of transaxles **1210***a* and **1210***b* respectively, accommodates any remaining driveline flexion that is not arrested by the mounting method detailed above. The output shaft **1007** of the prime mover mounted to vehicle frame **1202**, here internal combustion engine **1005**, can be seen in its proper alignment within central gear box **110**. By way of example, in a ZT mower application, v-belt **1206** may serve as the mowing deck drive belt.

FIGS. **38-40** depict the right side transaxle **1210***b* in greater detail. It should be understood that the following discussion is equally applicable to the left side transaxle **1210***a*, which will not be discussed further herein. Transaxle **1210***b* is housed in a three-part, two-chamber housing comprising central housing **1220**, central housing cover **1218** and gear housing cover **1224**. The housing sections are sealingly joined by a plurality of fasteners **1226**. Gear housing cover **1224** has an integral bearing support **1224***a*, which houses a bearing **1266** that provides rotational support for axle **1262** closer to its distal end, where wheel hub **1263** is mounted. (Axle **1262** is rotationally support at its proximal end by bearing **1264**, as best seen in FIG. **40**) Central housing **1220** and bearing support **1224***a* are both depicted with bosses having mounting holes **1223** and **1225**, respectively. It should be understood that other suitable locations for such mounting points may be selected depending on the vehicle application and the expected loading.

External to central housing **1220** are control arm **1250**, which regulates the output speed of the continuously variable toroidal drive mechanism **1230** located within central housing **1220**, and brake arm **1284**, which actuates a frictional brake mechanism that acts upon the planetary gear reduction mechanism **1267** located within gear housing cover **1224**. As previously stated, right side transaxle **1210***b* has two chambers formed when the three housing elements are secured together. A first chamber **1298** formed between the central housing cover **1218** and the central housing **1220** accommodates the continuously variable toroidal drive mechanism **1230**, and a second chamber **1299** formed between the central housing **1220** and the gear housing cover **1224** accommodates the planetary gear reduction mechanism **1267**. These two chambers are sealed from one another to prevent commingling of the distinct oils used in each chamber. Shaft seal **1296** serves this purpose. Unlike the typical oils or hydraulic fluids used in planetary reduction mechanisms, a specialized traction oil is utilized in a continuously variable toroidal drive mechanism to provide the necessary frictional engagement between the internal rollers and disks that produce the drive mechanism's variable output. Fill plug **1219** permits access to first chamber **1298** for purposes of traction oil addition. Fill plug **1222** permits access to the second chamber **1299** for addition of oil or hydraulic fluid to the planetary reduction mechanism **1267**.

While the mechanism **1230** depicted in FIG. **40** is depicted as a continuously variable toroidal drive mechanism it is to be understood that this is for exemplar purposes only. Any toroidal transmission mechanism capable of converting a drive input into continuously variable drive output is contemplated within the principles of the invention. Input shaft **1228***b*, which can be of variable length to accommodate a range of vehicle frame widths, receives its motive input from central output shaft **133***b*. Input shaft **1228***b* is rotationally supported on bearing **1260** and hollow shaft/pinion gear **1268** which runs on needle bearing **1261**. Input shaft **1228***b* provides rotational input to drive disk **1234** of the continuously variable toroidal drive mechanism **1230**. Drive disk **1234** provides rotational input to a plurality of traction rollers **1243**, which run upon a slidable bearing **1255**. Each traction roller **1243** rotates about an axial spindle **1245** that is supported by a roller arm **1247** at each end of the spindle **1245**. The roller arms **1247** track along curved surfaces within a retaining cage **1231** and along the sides of slidable bearing **1255**, permitting the traction roller **1243** to tilt on its spindle **1245**. Slidable bearing **1255** is supported on and rotates about a pair of bearing supports **1257** and **1259** which travel axially along input shaft **1228***b*. A slider block **1253** straddles slidable bearing **1255** to act on bearing supports **1257** and **1259** under the influence of a rotatable cam shaft **1251** that engages a slot on the slider block **1253**. Control arm **1250** is affixed to the external end of cam shaft **1251** to provide rotational control inputs. Rotation of control arm **1250** and cam shaft **1251** produces an axial movement of slider block **1253**, bearing supports **1257** and **1259**, and slidable bearing **1255**. The axial movement of slidable bearing **1255** within retaining cage **1231** causes traction rollers **1243** to tilt upon their spindles **1245**, changing the contact point between the traction rollers **1243**, drive disk **1234**, and an output disk **1238** affixed to hollow shaft/pinion gear **1268**. The change in contact point effectively changes the circumference of the traction roller **1243** experienced by drive disk **1234** and output disk **1238**, and thus changes the relative speeds experienced by drive disk **1234** and output disk **1238** to enable speed control. Drive disk **1234** and output disk **1238** turn at equivalent rates when traction rollers **1243** are not tilted on their spindles **1245** and track along the center of slidable bearing **1255**.

To maintain manageable control or axial forces at the traction interface between drive disk **1234**, traction rollers **1243**, and output disk **1238** under various loads, a spring washer **1235** is inserted to act upon drive disk **1234**. Correspondingly, to balance this axial loading, a thrust bearing **1239** and thrust washer **1241** are inserted between output disk **1238** and the wall of central housing **1220**. It should also be noted that retaining cage **1231**, which rests upon both drive disk **1234** and output disk **1238**, is prevented from rotating therewith by an anti-rotation bracket **1249**.

To accommodate the offset of input shaft **1228***b* and axle **1262**, and to provide a first stage of reduction, a rotatable ring gear **1270** is driven by hollow shaft/pinion gear **1268**. The rotatable ring gear **1270** is engaged to sun gear **1272** which rotates about axle **1262**. Sun gear **1272** drives a plurality of planet gears **1274** retained by a carrier **1276**. The planet gears **1274** run within a fixed ring gear **1278** to thereby rotate carrier **1276** affixed to axle **1262** and achieve a second stage of speed reduction. Thus, the resultant rotational speed of axle **1262** and hub **1263** is the result of continuously variable speed control and a two stage reduction.

Rotation of brake arm **1284** acts to rotate a cam shaft (not shown) which forces a brake puck (not shown) into frictional engagement with the exterior surface of rotatable ring gear **1270**. The brake as configured functions as a parking brake.

FIG. **41** depicts a central gear box **910** which is similar to central gear box **210**, but differs in that main housing **911** encases an electric clutch assembly **940** within sump **960**, i.e. a wet clutch design. Electric clutch assembly **940** has a field coil **940***a* which is energized by a power source (not shown) through electrical connector **940***b* (as shown in FIG. **42**). When field coil **940***a* is energized, armature **940***c* is magnetized and attracts clutch piston **946**. The stack-up of clutch plates **947** consists of one set that is captured by a gear form (not shown) native to outer clutch plate carrier **940***d* and an interwoven set that is captured by a gear form native to inner clutch plate carrier **940***e*. Inner clutch plate carrier **940***e* is engaged to and rotates with input shaft **921** under power from a prime mover, such as the prime mover **905** schematically depicted in FIG. **42**. Outer clutch plate carrier **940***d* is engaged to power take off output shaft **942**, to selectively rotate with it upon actuation of electric clutch assembly **940**. Clutch plates **947** are advantageously faced with a friction layer (not shown); for example, a nonwoven cellulosic with binder. The movement of the clutch plates **947** is restricted vertically by armature **940***c*. Thus, movement of clutch piston **946** toward armature **940***c* places clutch plates **947** into frictional engagement. As a result, power take off output shaft **942** is rotated synchronously with input shaft **921**. Power take off output shaft **942** is rotatably supported in main housing **911** by bearing **966**. It should be noted that shaft seal **967** is incorporated to prevent leakage of oil from sump **960**. A pulley **949** may be fixed to output shaft **942** by a washer **973** and fastener **974** to selectively drive an implement (not shown) by means of a drive belt (not shown).

FIG. **42** illustrates an alternative embodiment of a modular drive system **900** that incorporates central gear box **910** with a slight modification, the replacement of combination gear **929** with individual transmission and distribution gears **923**, **931**, respectively. Each of these gears is splined to jack shaft **924** to drive distribution gear set **930**. The configuration of distribution gear set **930**, which rotates a pair of central output shafts **933***a*, **933***b* in opposite senses, is the same as previously detailed for distribution gear set **530** and will not be further described herein. The central output shafts **933***a*, **933***b* are coupled to the input shafts **983**, **993** of the left and right side drive mechanisms **980**, **990**, respectively. In this fashion, a first shaft end is accessible from one side of the housing and a second shaft end is accessible from a second side of the housing opposite the first side, whereby motive force from prime mover **905** may be transferred to a pair of wheels **989**, **999** to propel and steer a vehicle so equipped.

It should be understood that the depicted configuration of distribution gear set **930** is illustrative only, and other known configurations, such as that of distribution gear set **230** where a single, central output shaft **233** is driven, may be utilized in a central gear box having an electric clutch disposed in its sump. And as similarly depicted in FIG. **2**, an intermediate shaft and coupling may be disposed between each of the input shafts **983**, **993** of the left and right side drive mechanisms **980**, **990**, respectively, and the corresponding central output shafts **993***a*, **993***b*.

While specific embodiments have been described in detail, it will be appreciated by those skilled in the art that various modifications, combinations and alternatives to those presented herein could be developed in light of the overall teachings of the disclosure. Accordingly, the particular arrangements disclosed are meant to be illustrative only and not limiting as to the scope of the invention which is to be given the full breadth of the appended claims and any equivalent thereof.

What is claimed is:

1. A power distribution assembly for use with an engine, the power distribution assembly comprising:
- a housing;
- an outer carrier that is rotatable and configured to be driven by an input shaft;
- one or more clutch plates secured to the outer carrier;
- one or more brake plates secured to the housing;
- an inner carrier that is rotatable and configured to occupy a first position where the inner carrier is engaged with the one or more clutch plates and a second position where the inner carrier is engaged with the one or more brake plates;
- a first washer engaging the one or more clutch plates;
- a second washer engaging the one or more brake plates;
- a retaining ring coupled to and extending from an outer surface of the inner carrier and disposed between the first washer and the second washer, wherein the retaining ring is configured to engage the first washer to cause the inner carrier to engage the one or more clutch plates in the first position, and wherein the retaining ring is configured to engage the second washer to cause the inner carrier to engage the one or more brake plates in the second position;
- a spring biasing the inner carrier away from the first position and toward the second position;
- a power take off output shaft driven by the inner carrier; and
- an actuator assembly having a first end disposed inside the housing and a second end disposed external to the housing, the actuator assembly configured to occupy a resting position and an active position, wherein in the resting position, the actuator assembly is disengaged from the inner carrier, and in the active position, the actuator assembly is engaged with the inner carrier to oppose a biasing force of the spring and retain the inner carrier in the first position.

2. The power distribution assembly of claim **1**, wherein the spring is disposed about the input shaft.

3. The power distribution assembly of claim **2**, wherein in the resting position, the actuator assembly enables the spring to retain the inner carrier in the second position.

4. The power distribution assembly of claim **1**, wherein the outer carrier and the one or more clutch plates are configured to rotate in unison with the input shaft.

5. The power distribution assembly of claim **4**, wherein the power take off output shaft is configured to rotate in unison with the inner carrier.

6. The power distribution assembly of claim **5**, wherein the inner carrier is configured to rotate in unison with the input shaft when occupying the first position.

7. The power distribution assembly of claim **6**, wherein the inner carrier is configured to arrest rotation of the power take off output shaft when occupying the second position.

8. The power distribution assembly of claim **1**, wherein the actuator assembly comprises a rotatable actuator handle, an actuator shaft affixed to the actuator handle, and an actuator fork affixed to the actuator shaft, wherein rotation of the actuator handle induces rotation of the actuator shaft, which induces rotation of the actuator fork, and wherein rotation of the actuator fork applies a counter force against the inner carrier.

9. The power distribution assembly of claim **8**, wherein the applied counter force opposes the biasing force of the spring and motivates the inner carrier toward the first position.

10. The power distribution assembly of claim **9**, wherein the actuator assembly further comprises a thrust bearing disposed between the actuator fork and the inner carrier such that the counter force is applied against the inner carrier at least via the thrust bearing.

11. The power distribution assembly of claim **10**, wherein when the actuator assembly occupies the active position, the thrust bearing directly contacts the inner carrier.

12. The power distribution assembly of claim **10**, wherein the thrust bearing is disposed about the power take off output shaft.

13. The power distribution assembly of claim **1**, further comprising a transmission gear splined to the input shaft above the outer carrier.

14. The power distribution assembly of claim **1**, wherein the outer surface of the inner carrier defines a slot in which the retaining ring is received to couple to the inner carrier.

15. A power distribution assembly for use with an engine, the power distribution assembly comprising:
- a housing;
- a rotatable outer carrier configured to be driven by an input shaft;
- one or more clutch plates secured to the rotatable outer carrier such that the one or more clutch plates are configured to be driven by the input shaft;
- one or more brake plates secured to the housing;
- a rotatable inner carrier configured to occupy a first position to engage the one or more clutch plates and a second position to engage the one or more brake plates;
- a first washer engaging the one or more clutch plates;
- a second washer engaging the one or more brake plates;
- a retaining ring coupled to and extending from an outer surface of the rotatable inner carrier and disposed between the first washer and the second washer, wherein the retaining ring is configured to contact the first washer to cause the rotatable inner carrier to engage the one or more clutch plates in the first position, and wherein the retaining ring is configured to contact the second washer to cause the rotatable inner carrier to engage the one or more brake plates in the second position;
- a spring disposed about the input shaft and biasing the rotatable inner carrier away from the first position and toward the second position; and
- a power take off output shaft driven by the rotatable inner carrier and rotatably supported in the housing, wherein the input shaft is at least partially supported by the power take off output shaft.

16. The power distribution assembly of claim **15**, further comprising an actuator assembly having a first end disposed inside the housing and a second end disposed external to the housing.

17. The power distribution assembly of claim **15**, further comprising a bushing disposed between the input shaft and the power take off output shaft.

18. The power distribution assembly of claim **16**, wherein the actuator assembly is configured to occupy a resting position and an active position.

19. The power distribution assembly of claim **18**, wherein, in the resting position, the actuator assembly configured to be disengaged from the rotatable inner carrier.

20. The power distribution assembly of claim **18**, wherein, in the active position, the actuator assembly is configured to be engaged with the rotatable inner carrier to oppose a biasing force of the spring to retain the rotatable inner carrier in the first position.

21. The power distribution assembly of claim **15**, wherein the outer surface of the inner carrier defines a slot in which the retaining ring is received to couple to the rotatable inner carrier.

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